Technical Report

Rapport technique

1984/11

TR-LT-010
NRC NO. 24201

TESTING OF WATER-TO-REFRIGERANT EVAPORATORS FOR RESIDENTIAL HEAT PUMP SYSTEMS

P.J. Hawken, P.F. Hearty

Division of
Mechanical Engineering

Division de
génie mécanique

National Research Council Canada

Conseil national de recherches Canada

Canada

85-4903/

DIVISION OF MECHANICAL ENGINEERING PUBLICATIONS

DM **(Division of Mechanical Engineering Report)**
Scientific and technical information considered important, complete and a lasting contribution to existing knowledge.

TR **(Technical Report)**
Information less broad in scope but a substantial contribution to existing knowledge.

CTR **(Controlled/Classified Technical Report)**
A Technical Report with controlled distribution for national security, proprietary or other reasons.

LM **(Laboratory Memorandum)**
Preliminary or exploratory information with controlled distribution.

CAT **(Calibration Analysis and Test Report)**
Information on minor laboratory projects or services.

PUBLICATIONS DE LA DIVISION DE GÉNIE MÉCANIQUE

DM **(Rapport de la Division de génie mécanique)**
Informations scientifiques et techniques jugées importantes, complètes et susceptibles de contribuer de façon durable à l'avancement des connaissances courantes.

TR **(Rapport technique)**
Informations de moindre importance, mais pouvant contribuer substantiellement à l'avancement des connaissances actuelles.

CTR **(Rapport technique à diffusion contrôlée/classifiée)**
Rapport technique à diffusion contrôlée pour des raisons de sécurité nationale, de propriété intellectuelle et autres.

LM **(Mémoire de laboratoire)**
Informations préliminaires ou de nature exploratoire à diffusion contrôlée.

CAT **(Rapport d'étalonnage d'analyse et d'essai)**
Informations sur de petits projets ou des services de laboratoire.

UNLIMITED
UNCLASSIFIED

ILLIMITÉ
NON CLASSIFIÉ

TESTING OF WATER-TO-REFRIGERANT EVAPORATORS FOR RESIDENTIAL HEAT PUMP SYSTEMS

ESSAI D'ÉVAPORATEURS EAU-RÉFRIGÉRANT POUR LES INSTALLATIONS DE POMPE À CHALEUR POUR LES HABITATIONS

P. J. Hawken
P. F. Hearty

Technical Report

1984/11

Rapport technique

TR-LT-010
NRC NO. 24201

T. R. Ringer, Head/Chef
Low Temperature Laboratory/
Laboratoire des basses températures

E. H. Dudgeon
Director/
Directeur

ABSTRACT

Eight water-to-refrigerant evaporators suitable for residential heat pump use were tested under similar conditions. The test results showed a great variation between the performance of individual evaporators. Performance of the evaporators was not linked to size, weight or cost but rather to attention to details of design and fabrication. The effect of evaporator performance on the heat pump system is discussed.

RÉSUMÉ

Huit évaporateurs eau-réfrigérant appropriés pour les installations de pompe à chaleur pour les habitations ont été mis à l'essai dans des conditions similaires. Les résultats des essais démontrent un écart considérable dans le rendement des évaporateurs individuels. Le rendement des évaporateurs n'est pas relié à la taille, au poids ni au coût, plutôt à l'attention accordée aux détails de conception et de fabrication. Le rapport traite également des effets du rendement de l'évaporateur sur une installation de pompe à chaleur.

CONTENTS

		Page
	ABSTRACT	(iii)
	RÉSUMÉ	(iii)
	LIST OF ILLUSTRATIONS	(iv)
	LIST OF TABLES	(v)
	APPENDICES	(v)
1.0	INTRODUCTION	1
2.0	TEST APPARATUS	1
3.0	TEST PROCEDURE	3
4.0	TEST RESULTS AND DISCUSSION	5
	4.1 General	5
	4.2 Flooded Evaporator (No. 8)	6
	4.3 Thermal Performance Comparison	7
	4.4 Refrigerant Side Pressure Loss	8
	4.5 Effect of Evaporator Orientation	9
5.0	CONCLUSIONS	10
6.0	REFERENCES	11
	DOCUMENTATION PAGE	

LIST OF ILLUSTRATIONS

Figure		
1	Schematic of Test Rig	12
2	Refrigerant Pressure-Enthalpy Diagram Showing Operating Conditions for Test	13
3	Fluid Temperature Profiles for a Counter Flow, Coaxial Tube-in-Tube Evaporator	14
4	Details of Evaporators Tested	15-31
5	Water Consumption for Evaporators	32
6	Effectiveness of Evaporators	33
7	Refrigerant Circuit Pressure Loss for Evaporators	34

CONTENTS (Cont'd)

LIST OF ILLUSTRATIONS (Cont'd)

Figure		Page
8	Effect of Water Flow Rate through the 'Flooded' Type Evaporator (No. 8) on the Compressor Can Temperature	35
9	Effect of High Refrigerant Circuit Pressure Drop in the Evaporator with Fixed Compressor Suction Pressure	36
10	Effect of Evaporating Pressure on Performance of a Typical Hermetic, Reciprocating Compressor	37
11	Postulated Refrigerant Flow Pattern in Correctly Sized and Oversized Coaxial Tube-in-Tube Evaporators	38

LIST OF TABLES

Table		
1	Details of Evaporators Tested	39
2	Summary of Evaporator Test Results	40
3	Effect of Water Flow Rate on Heat Pump System Performance	41

APPENDICES

Appendix		
A	Summary of Instrumentation Used	A1
B	Summary of Evaporator Performance Data used in this Report	B1
C	Effect of Water Flow Rate on Heat Pump System Performance	C1

TESTING OF WATER-TO-REFRIGERANT EVAPORATORS FOR RESIDENTIAL HEAT PUMP SYSTEMS

1.0 INTRODUCTION

Considerable interest has been shown recently in water source heat pumps. Inherently, water-source heat pumps offer several important advantages over air-source heat pumps. Primarily, the water, either ground water or surface water, is usually warmer than the ambient air during Canadian winters. Therefore the water-source heat pump will operate at a higher coefficient of performance (C.O.P.). Secondly, water is a much better heat transfer fluid than air. Hence, the evaporator on a water-source heat pump is typically smaller and less expensive (for volume production) than the evaporator on an air-source heat pump.

Canada, with its large and ubiquitous water resources, has a significant potential market for water-source heat pumps. Consequently the Low Temperature Laboratory of the Division of Mechanical Engineering, as part of its overall heat pump programme, has been investigating the design of water-source heat pumps.

One of the major deficiencies found was in the design of the evaporator. Typically, commercial water-source heat pumps have been using evaporators that were originally designed as water-cooled condensers. These condensers, when used as evaporators, usually exhibit a high refrigerant pressure drop and hence degrade the overall performance of the heat pump. Also, condensers were not designed to resist damage caused by water freezing in the flow passages, a strong possibility with the low temperature waters found in Canada.

Many residential-sized water-source heat pumps operate from marginal wells of low output and/or considerable depth. Therefore it is important that the evaporator be as efficient as is economical and use the least amount of water.

In support of the fledgling water-source heat pump industry in Canada, the task of evaluating commercially available water-source evaporators was undertaken by the Laboratory. This report describes the test apparatus constructed, the test procedure, and results from the evaporators tested. The capability of the evaporators to undergo 'freeze-up' without damage was not investigated.

2.0 TEST APPARATUS

Most of the residential-sized heat pumps commercially produced at present have a heating capacity in the range 7 to 14 kW and use R22 refrigerant. Therefore a test apparatus was constructed to simulate performance of a heat pump of this size [Fig. 1]. A condensing unit (of nominal 11 kW size) from a commercially manufactured air conditioner was used. This condensing unit came complete with hermetic compressor, compressor protection devices and

an air-cooled condenser. A water-cooled condenser was substituted for the air-cooled condenser to better control the condenser conditions.

After much trial and error, an electric expansion valve was found to give the best refrigerant control. The valve's sensor was disconnected and the valve controlled from an adjustable voltage source. In this way, any desired setting of the valve could be maintained and reproduced at a later date, if need be, by setting the same voltage on the voltage source.

Water flow to the evaporator was measured using a calibrated rotameter. Temperatures of water into and out of the evaporator were initially measured with a quartz thermometer of high accuracy and resolution (± 0.01°C). Unfortunately, this instrument failed during the testing and an RTD unit with similar accuracy was substituted. From the water flow rate and temperature difference across the heat exchanger, the energy lost by the water could be calculated. Water was supplied from a 2300-litre storage tank, the temperature of which was controlled by a 180-litre residential water heater with a 6 kW electric resistance heater and a 3 kW in-line electric resistance heater. Power to the 6 kW heater was switched on or off as required and an adjustable power source was used to vary power to the 3 kW heater. Pressure drop across the water side of the evaporator was measured using a bourdon type pressure gauge.

Refrigerant flow was measured in the sub-cooled liquid refrigerant line, upstream of the expansion valve. Two different flow meters in series were used here to ensure a high confidence level in the measurement. The first was a turbine meter which provides a measure of the volumetric flow rate. From the pressure and temperature of the liquid, the refrigerant density can be calculated and a mass flow rate obtained. A mass flow meter, operating on the vibrating 'U' tube principle, was also used. Care was taken that the liquid refrigerant was always subcooled by at least 5°C to ensure there were no vapour bubbles present in the meters and no cavitation was occurring in the turbine meter. The mass flow meter was used as the primary source for the mass flow rate; however, agreement between the two meters was consistently about 3.5%.

Refrigerant temperatures around the circuit were obtained using type T thermocouples soldered to the copper pipes and then insulated. Pressures were obtained using variable capacitance type pressure transducers.

Data from transducers with electrical output were collected using a high quality data acquisition system integrated with a small desktop computer. Details of the instrumentation used are given in Appendix A.

3.0 TEST PROCEDURE

The test procedure was devised to provide a common operating condition which would be representative of a water-source heat pump in a typical residence. Therefore, the following parameters were kept constant for all the evaporators tested:

(1)	water inlet temperature	10°C
(2)	energy lost by the water	6.66 kW
(3)	temperature of sub-cooled liquid refrigerant to the expansion valve (corresponding to an inlet refrigerant enthalpy of 98 kJ/kg)	43.3°C
(4)	refrigerant pressure at the evaporator outlet of (corresponding to a saturated refrigerant temperature of -1.7°C)	470 kPa

In addition to the above conditions, it was attempted to keep the superheat of the refrigerant out of the evaporator at a constant 5.8°C. Unfortunately, for the most inefficient coaxial tube-in-tube evaporators tested, the 1.1 kW centrifugal water pump was not sufficiently large to pump enough water to obtain the degree of superheat required, and consequently a lower degree of superheat had to be accepted for these evaporators. One of the eight evaporators tested was of the flooded type. Therefore, because of the inherent nature of this evaporator, only saturated refrigerant vapour could be obtained at the outlet. The evaporation process and test conditions are indicated on the pressure-enthalpy diagram for R22 in Figure 2.

The superheat of 5.8°C was chosen for two reasons. Firstly, the heat pump system coefficient of performance (COP) is affected by the superheat of vapour to the compressor, and recent work [Ref. 1] indicates that the maximum system COP occurs for a range of superheats from 5°C to 8°C for R12. This conveniently represents a range of superheats that can be attained using conventional expansion devices. A similar situation was assumed to exist for R22 heat pump systems. Secondly, from Figure 3 [taken from Ref.2] it is apparent that superheating of the refrigerant consumes an excessive amount of heat exchanger surface relative to the energy obtained. Therefore the 5.8°C superheat figure was selected as a compromise between obtaining the best heat pump system COP and maximizing the use of evaporator heat exchange surface.

The test of an evaporator entailed placing the evaporator in the test rig, making the necessary connections, evacuating the refrigerant side of the evaporator and insulating the whole evaporator with about 100 mm of fibreglass insulation blanket. The fibreglass blanket was wrapped in a covering of PVC film to prevent any splashing of water or condensed moisture from degrading the

insulation. After starting up the heat pump system, the refrigerant charge was adjusted and the water heaters in the water tank were left off while the heat pump lowered the tank temperature to the desired 10°C. When the water temperature reached 10°C, the electric heaters were switched on and the power level adjusted with the variable power supply to maintain the water temperature.

The expansion valve, condenser and evaporator water flow rates and refrigerant charge were adjusted by trial and error to obtain the desired conditions. This process was time consuming, but after some experience it could be accomplished within a morning's work. Readings were only taken after steady state conditions were achieved; typically such that 30 minutes of operation had elapsed since the last minor alteration to the system.

Capacity of the evaporator can be calculated by the familiar heat exchanger effectiveness concept, i.e.

$$Q = \varepsilon \, m_w C_{pw} (T_{w_{in}} - T_{R_{in}}) \qquad (1)$$

$$= UA \, \Delta T$$

where: Q = rate of heat transfer (Watts)
ε = heat exchanger effectiveness (dimensionless)
m_w = mass flow rate of water (kg/s)
C_{pw} = specific heat of water (J/kg-°C)
$T_{w_{in}}$ = temperature of water into evaporator (°C)
$T_{R_{in}}$ = temperature of refrigerant into evaporator (°C)
UA = overall conductance of the heat exchanger (W/°C)
ΔT = average temperature difference between the fluid streams (°C).

Figure 3 shows the fluid temperature profiles for a typical counter flow evaporator such as the coaxial tube-in-tube models tested. There are two flow sections apparent from this figure; a two-phase region with almost constant temperature boiling refrigerant flow and a single-phase region with the refrigerant being superheated. Because of the different pressure drops and flow passage geometries encountered in the evaporators tested, it was not practical to determine the temperature of the refrigerant between each of the two flow sections. Therefore it was not possible, for these experiments, to examine the performance of the separate flow sections.

For constant Q, C_{pw}, $T_{w_{in}}$ and $T_{R_{in}}$ the overall effective-

ness of the evaporator is inversely proportional to the water flow rate, i.e.:

$$\varepsilon \propto \frac{1}{\dot{m}_w} \qquad (2)$$

In this way it is seen that the better evaportors will use less water than the poorer ones. Unfortunately, because each evaporator had a different refrigerant pressure drop (40-130 kPa) and only the outlet pressure could be held constant, the inlet pressure (and hence saturated temperature) varied from evaporator to evaporator. Therefore the relationship in (2) was not always strictly true, i.e. the effectiveness was also a function of refrigerant temperature through the evaporator.

To simplify the analysis it was decided to use an overall effectiveness factor as defined by (1). Each evaporator was therefore compared on the basis of transferring a specified heat rate (6.66 kW) from 10°C water with a specified refrigerant inlet enthalpy (98 kJ/kg) and outlet pressure (470 kPa). For the non-flooded evaporators, a superheat of 5.8°C was maintained for the exiting vapour.

Eight different, commercially available evaporators from five different manufacturers were tested. Details of their construction are given in Figure 4 and Table 1. Evaporator No. 5 was a prototype of a new model on loan from a Canadian heat pump manufacturer. It was not possible to section this evaporator or show its construction (which was very similar to evaporator No. 2).

4.0 TEST RESULTS AND DISCUSSION

4.1 General

The test data and calculated results are given in Appendix B and a summary of the test results is given in Table 2. Figures 5 and 6 show the water consumption and heat exchanger effectiveness plotted against the mass of the evaporator, while Figure 7 shows the refrigerant pressure drop for each evaporator plotted against the mass of the evaporator.

Note that evaporators No. 4 and 5 were operated with a lower superheat than the specified 5.8°C (and a lower capacity for No. 4). This was because the test rig could not pump sufficient water through these evaporators to achieve the specified superheat. Therefore the performance of these two evaporators is overstated by these figures, i.e. if they had been operated at the 6.66 kW capacity with 5.8°C superheat, much more water would have been required. Also, if evaporator No. 4 had been operated at its rated capacity, the refrigerant flow rate would have been greater and hence the refrigerant side pressure drop would be much greater.

4.2 Flooded Evaporator (No. 8)

Evaporator No. 8 is of the combination flooded evaporator and suction accumulator type and hence only produces saturated vapour. Because of this it was not possible for the single-speed compressor to attain the desired capacity without lowering the suction pressure slightly. There was also a certain arbitrariness associated with the choice of operating point for this evaporator. By adjusting the refrigerant charge the same capacity could be attained with a range of water flow rates from 0.19 to 0.66 kg/s [see Figure 8]. Referring to the details of this evaporator in Figure 4.8, this behaviour was explained in the following way:

The evaporator is operated in a vertical orientation. At high water flow rates the overall heat transfer rate across the tubes was high and only a relatively small fraction of the tube surface area was needed to boil off the required amount of refrigerant. Therefore most of the evaporator shell was filled with vapour and the boiling liquid refrigerant level was well down in the evaporator can. As the water flow rate decreased, the heat transfer coefficient also decreased, and so more tube area was required to evaporate off the required amount of refrigerant. Therefore the liquid refrigerant level in the evaporator can rose as the water flow rate was reduced. It should be remembered that the refrigerant charge was adjusted at each water flow rate to obtain the desired steady state conditions. For water flow rates greater than about 0.275 kg/s, the vapour leaving the saturated evaporator was essentially dry with a small amount of liquid refrigerant introduced via the orifice in the bottom of the large central tube. This sub-system served to introduce oil-rich liquid refrigerant to the vapour going to the compressor, i.e. to act as a compressor lubricating oil return. The amount of liquid refrigerant in the suction line (introduced via the small orifice in the large central tube) increased gradually as the 'head' of liquid refrigerant above it increased; as evidenced [in figure 8] by the slight slope of the curve at water flow rates greater than 0.27 kg/s. Liquid refrigerant introduced in small amounts into the compressor cools the compressor as the refrigerant evaporates; hence the more refrigerant that is introduced, the cooler the compressor runs. Only a very limited amount of liquid refrigerant can be introduced before mechanical damage to the compressor will result.

As the water flow rate was reduced below 0.27 kg/s, the boiling liquid refrigerant rose to such levels that drops of liquid refrigerant, thrown up by the violent boiling action in the evaporator shell, could enter the suction opening in the large central tube at the top of the shell. This occurrence was evidenced by a sharp decrease in the compressor can temperature as more liquid refrigerant was sucked in. During this stage the compressor was noticed to operate much more noisily. Therefore, to prevent possible damage to the compressor, the operating water flow rate for this evaporator was (somewhat optimistically) chosen as 0.274 kg/s; i.e. on the 'knee' of the curve in Figure 8.

4.3 Thermal Performance Comparison

Figures 5 and 6 show that generally the effectiveness is almost inversely proportional to the water flow rate required, i.e. the better evaporators use the least amount of water for the same duty. A wide range of effectiveness (0.18 to 0.91) and water flow rate (1.077 to 0.204 kg/s) was encountered. Therefore a heat pump system which uses the poorest evaporator will have to pump five times the water which a system with the best evaporator would use. It is interesting to note that there is no direct correlation between performance and the type of construction or mass of evaporator. The best and the worst were of essentially the same construction (coaxial tube-in-tube) and the two heaviest evaporators performed very poorly.

The severity of the penalty incurred by the heat pump system in using a poor evaporator is illustrated by the figures in Table 3 which are based on the example described in Appendix C. This example uses a typical case where a water-source heat pump for a residence is supplied with water from a well with an electric submersible well pump. With a 'good' evaporator which uses only 0.204 kg/s of water, the steady state system COP is 2.79, but with a 'poor' evaporator which uses 1.077 kg/s, the system COP falls to 2.38; a drastic and unnecessary degradation of system performance. Therefore it is apparent that the correct choice of an evaporator is of great importance in the overall design of a water-source heat pump.

It was very difficult to obtain cost figures for each of the evaporators so that they could be compared on the same basis; i.e. prices vary depending on the quantity purchased, the purchaser (e.g. original equipment manufacture (OEM) or retail), the seller (e.g. manufacturer or agent) and the time of purchase (e.g. the agent may be trying to reduce his inventory). Given that price comparisons may not be valid in all cases, Table 1 contains relative costs for the evaporators tested, based on the OEM cost for small lots. From these costs it is obvious that the better evaporators are not necessarily the most expensive; in fact the second best evaporator was also the cheapest by far. An examination of the relative costs per unit weight of evaporator reveals a wide variety in the pricing structure. It should be noted that evaporator No. 7 is of all stainless steel construction, whereas all the others have a steel and copper construction; hence the high cost per unit weight of No. 7. This high cost may be worth paying if a heat source fluid is used which is corrosive to copper. Also, evaporator No. 8 was of tube-in-shell construction and hence its fabrication costs would probably be higher than the coaxial tube-in-tube evaporators.

Some interesting conclusions arise if a comparison is made of the performance of, say, evaporator No. 1 and evaporator No. 2. Both evaporators are of a similar type in that they both use a coaxial tube-in-tube construction with water in the inner copper tube and refrigerant in the annulus between the inner copper

tube and outer steel tube. The main difference in construction between the two lies in the fact that evaporator No. 1 uses a plain copper tube with small 'fins' soldered to the outside of the tube, whereas evaporator No. 2 uses a convoluted copper tube with a mechanically roughened (sandblasted) outer surface. Therefore, evaporator No. 1 applies heat transfer enhancement to the refrigerant side with its associated high transfer film coefficient and none to the water side, whereas evaporator No. 2 applies heat transfer enhancement to both refrigerant and water sides. The result of this is that evaporator No. 2 far outperforms evaporator No. 1, see Table 2, and with one third less material. Clearly then it is important to enhance the heat transfer coefficient on the water side as well as the refrigerant side; and the convoluted tube offers an inexpensive and efficient means of achieving this.

4.4 Refrigerant Side Pressure Loss

The refrigerant circuit pressure loss experienced by each evaporator did not appear to follow any pattern either. Evaporators 5, 2 and 3 which showed the best heat transfer effectiveness were in the middle of the range for refrigerant pressure drop, as was evaporator No. 6 with the worst effectiveness [see Figure 7]. The mass of the evaporator did not seem to have a major effect on the refrigerant pressure loss either. The two lightest evaporators (4 and 7) had the greatest and least pressure drops respectively, while the heaviest three evaporators (1, 6 and 8) demonstrated a range of pressure drops from low to average.

All of the evaporators demonstrated refrigerant side pressure drops which are considered to be too high. Typically an evaporator should not have a refrigerant side pressure drop greater than about 30 kPa. Some of the evaporato s tested had pressure drops as high as 128 kPa. These high pressure drops will degrade the performance of the heat pump system. For a fixed evaporator outlet pressure, the high pressure drop evaporator will have a higher average temperature than the low pressure drop evaporator [see Figure 10]. This means there will be less potential temperature difference between the two fluid streams and so to transfer the same amount of heat the heat exchanger has to have more surface area or a higher overall heat transfer coefficient (which is often achieved by flowing more water).

Should the evaporator outlet pressure not be fixed, the compressor will pull the evaporator pressure down until the new average evaporator temperature establishes a new equilibrium. Therefore a higher pressure drop evaporator will cause the system to operate at a lower pressure on the suction side of the compressor. Figure 11 shows the effect on a typical compressor of reducing the suction pressure, while keeping the condenser conditions the same. A drop of only 50 kPa in suction pressure causes the COP to drop by 0.23 and the capacity to drop by 1.3 kW (16%).

Some of the excessive refrigerant pressure drop in the evaporators was due to poorly assembled end fittings. For example,

the photograph of sectioned evaporator No. 3 in Figure 4.3 shows how the end of the tube has been pushed down, away from the end socket, and hence has pinched off some of the refrigerant passage.

Evaporator No. 6 was an oversized unit designed for higher capacities (nominal 14 kW). Despite the increased heat exchange surface area, this unit had a low effectiveness which was attributed to the low water and refrigerant mass flow rates relative to the size of the flow passages, i.e. low Reynolds number flows. It was also postulated that the low refrigerant flow rates in the large annulus resulted in the liquid refrigerant flowing in a stream along the bottom of the annulus (the evaporator coils were tested in the horizontal position) and hence not coming into effective contact with much of the inner tube carrying the water [see Figure 11]. Surprisingly, the refrigerant side pressure loss for this evaporator was not as low as would be expected; it lay near the middle of the range of evaporator pressure drops in Figure 7.

4.5 Effect of Evaporator Orientation

Most commercially available water source heat pumps with coaxial tube-in-tube evaporators operate with the tube coils in a vertical plane. The reason for this appears to be primarily for the convenience of packaging the unit. Tests were conducted to determine if any penalty was incurred by orienting this type of evaporator in a vertical as opposed to a horizontal position. Evaporator No. 2 was tested under identical conditions in both a horizontal and vertical position; a summary of the test results is given in tables B3 and B4. The performance of the evaporator was very similar in both positions, with the vertical position tests showing slightly better performance. For example, the evaporator in the vertical position used 8.5% less water than in the horizontal position, though it also transferred 1.5% less heat. Although considerable effort was expended to make the conditions exactly the same for both orientations, there are slight differences which may contribute to some of the variation in performance between the two orientations.

Differences in performance between the two orientations are explained on the basis of refrigerant flow patterns. The flow pattern in the horizontal case would be similar to that in Figure 11a for much of the length of the evaporator. In the vertical case the flow pattern will not be as uniform along the evaporator. Due to the effect of gravity there will be a tendency for more liquid refrigerant to be present in the lower parts of the evaporator coil loops than in the upper part of the loops. Therefore the thickness of the liquid refrigerant film on the heated inner tube surface will vary not only in a radial direction but also along the length of each coil. Indications are that the boiling heat transfer rate greatly increases as the thickness of the liquid film decreases, until dryout of the liquid from the surface occurs [Ref. 3]. This would mean an increase in the heat transfer for the upper parts of the loop and a decrease in the heat transfer for the lower parts. This phenomenon combined with other factors such as increased

turbulence at the gas/liquid interface could be responsible for the slightly increased performance of the vertically oriented evaporator.

Time and resources permitted the testing of only one evaporator in the vertical orientation. Other evaporators may show different behaviour. For example, the larger refrigerant flow passages in evaporator No. 6 may result in a refrigerant velocity that is not high enough to continuously carry the liquid refrigerant over the top of each coil loop. Therefore liquid refrigerant could collect in the bottom of each loop until the back pressure and vapour velocity are great enough to push the slug of liquid over the 'top' of the coil loop and into the bottom of the next loop. This behaviour would cause the heated inner tube at the top of each loop to be alternately dry and flooded with liquid refrigerant, resulting in poor overall performance of the evaporator in the vertical position.

5.0 CONCLUSIONS

The choice of evaporator can have a major impact on the performance of a water-source heat pump. A poor evaporator which requires a large water flow rate will degrade the overall system performance due to the excessive water pumping power requirement. An evaporator with a high refrigerant side pressure drop will also degrade the system performance. In a system where the compressor suction pressure is not fixed, the higher refrigerant pressure drop associated with a poor evaporator will lower the compressor suction pressure, resulting in a lower system COP. For a system with a fixed suction pressure, the temperatures between the two fluid streams will be closer in a high pressure drop evaporator; and therefore more water flow rate will typically be needed to compensate.

A wide variation was found in the performance of the commercially available evaporators tested. The best evaporators were not necessarily the most expensive or the ones with the most material in them. Even evaporators of very similar construction performed quite differently. It is therefore concluded that careful design and fabrication with attention to details is very important in producing an evaporator that performs well. The coaxial tube-in-tube evaporators No. 2, No. 3 and No. 5 performed very well at the duty selected and were inexpensively priced.

These better evaporators are approaching the maximum effectiveness that is desirable for an evaporator in this application. For example, the bulk temperature of the water leaving evaporator No. 5 was 2.3°C. Any further reduction in outlet temperature may result in freezing up of the evaporator, a situation which could easily be triggered by a change in water inlet temperature or a reduction in the condensing temperature.

The performance of the co-axial tube-in-tube evaporators was greatly improved by heat transfer enhancement of the water side

as well as the refrigerant side. The convoluted tube offers an inexpensive and efficient way of achieving this enhancement. Mechanical roughening of the tube wall (e.g. by sandblasting) can also prove to be an economical way of enhancing the fluid heat transfer coefficient.

Evaporator No. 2 was tested in both the horizontal and vertical positions. This evaporator was found to perform slightly better in the vertical position. Other evaporators may not display similar behaviour.

6.0 REFERENCES

1. Hughes, D.W.
   McMullan, J.T.
   Mawhinney, K.A.
   Morgan, R.
   "Experimental Investigation of the Influence of Lubricating Oil on Heat Pump Performance".
   International Journal of Energy Research, Vol. 8, 1984, pp. 213-222.

2. Hawken, P.J.
   "An Analysis of a Coaxial, Tube-in-Tube, Water Heated Evaporator".
   National Research Council of Canada, unpublished report.

3. Parizhskiy, O.V.
   Chepurnenko, V.P.
   Lagota, L.F.
   Taranets, L.F.
   "Study of Boiling Heat Transfer with a Falling Film of Refrigerant".
   Heat Transfer - Soviet Research, Vol. 4, No. 4, July-Aug. 1972, pp.43-47.

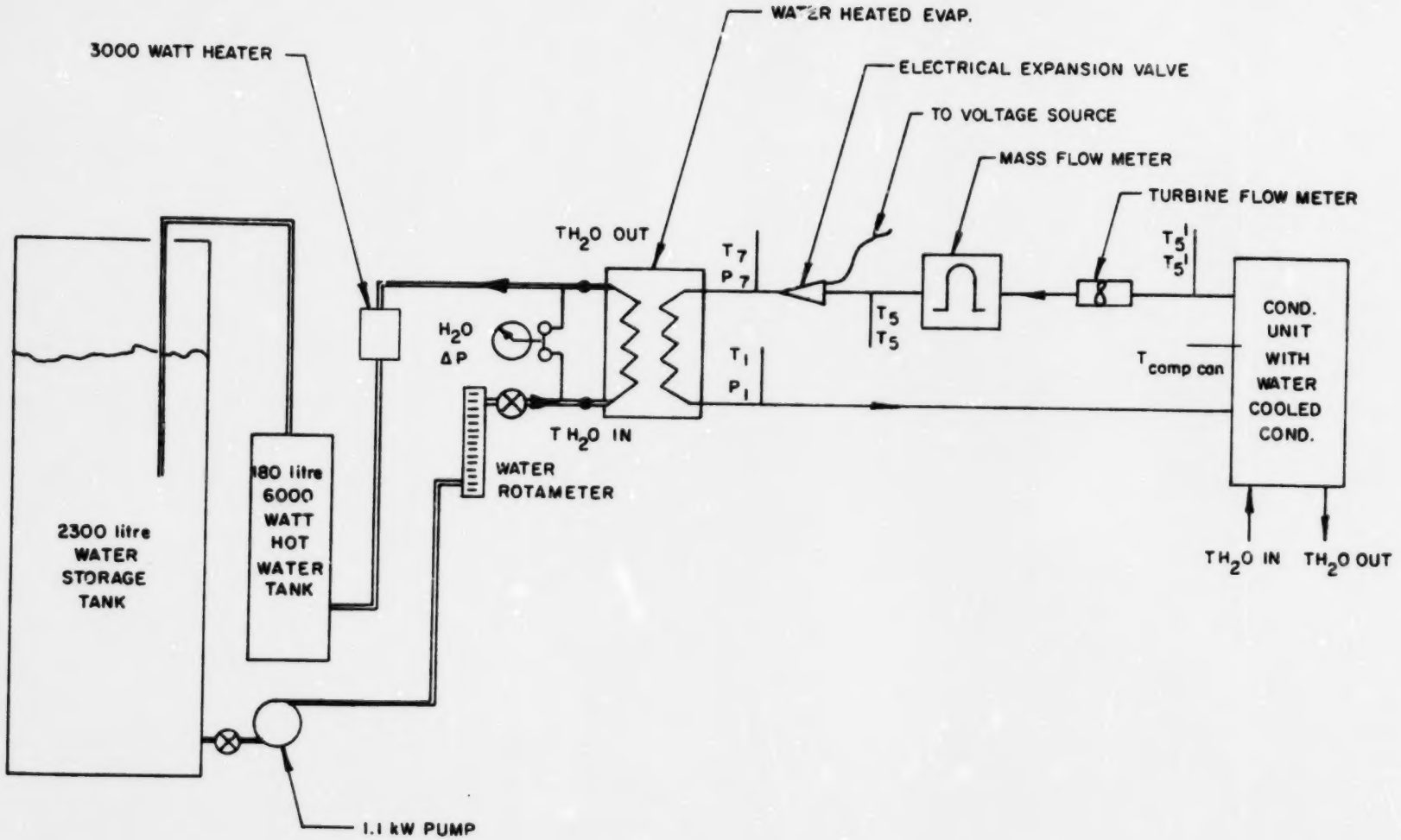


FIG. 1 SCHEMATIC OF TEST RIG

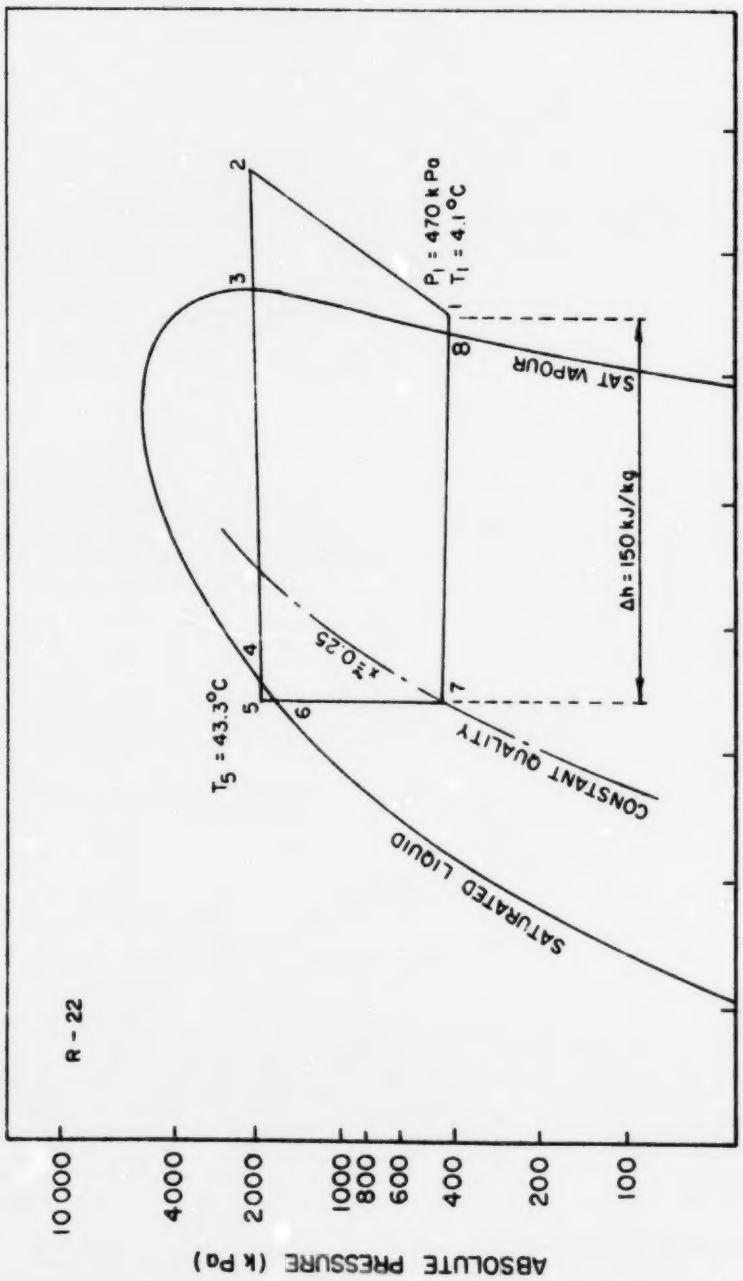


FIG. 2 REFRIGERANT PRESSURE - ENTHALPY DIAGRAM SHOWING OPERATING CONDITIONS FOR TEST

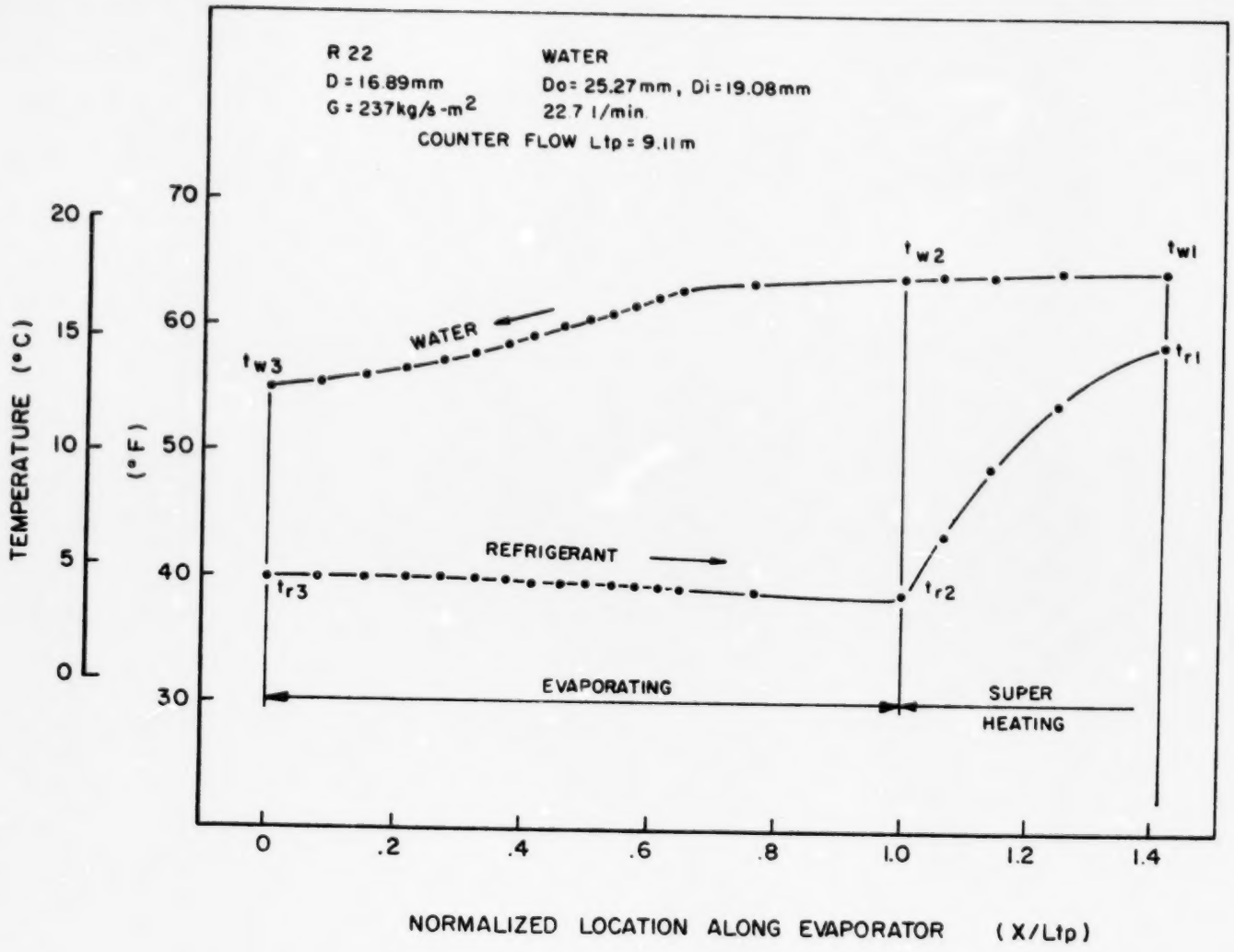


FIG. 3 FLUID TEMPERATURE PROFILES FOR A COUNTER FLOW, COAXIAL TUBE-IN-TUBE EVAPORATOR

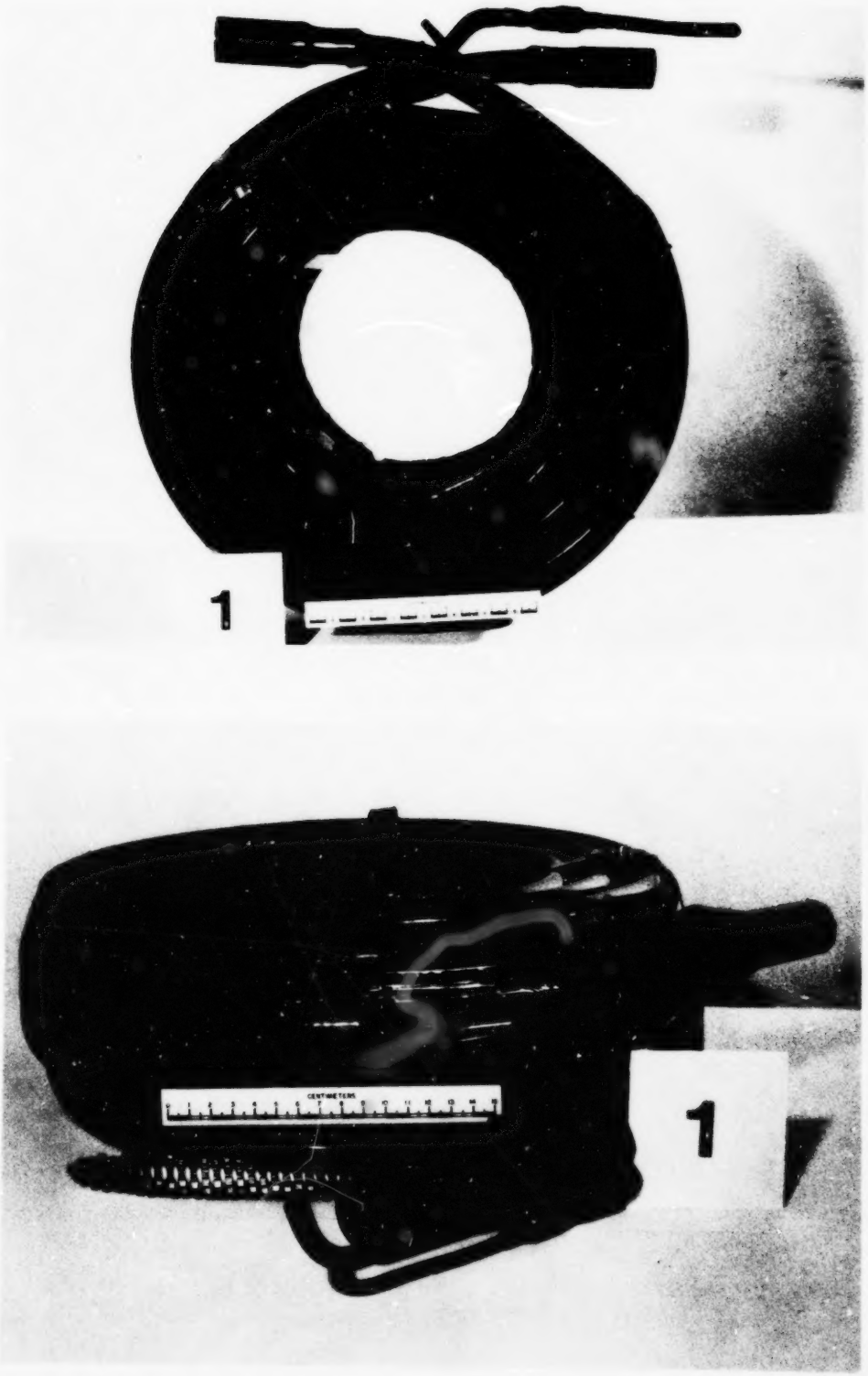
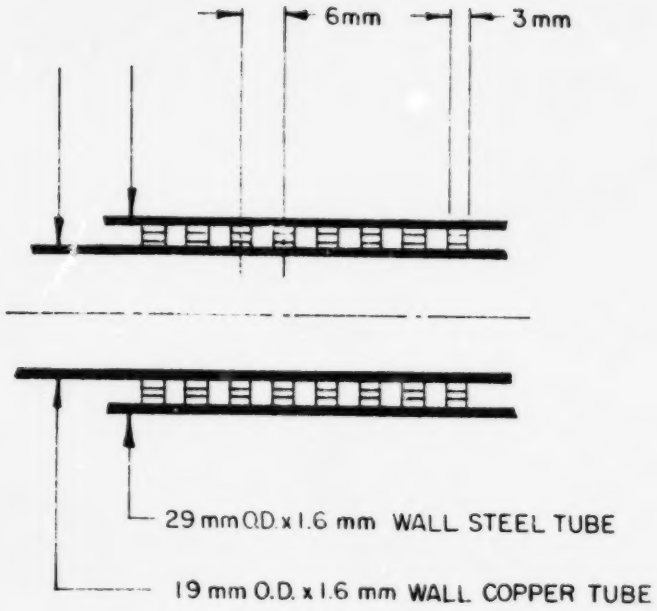


FIG. 4.1a EVAPORATOR NO. 1



APPROX. 10.2 m OF EACH SIZE OF TUBE

FIG. 4.1b DETAIL OF EVAPORATOR NO. 1

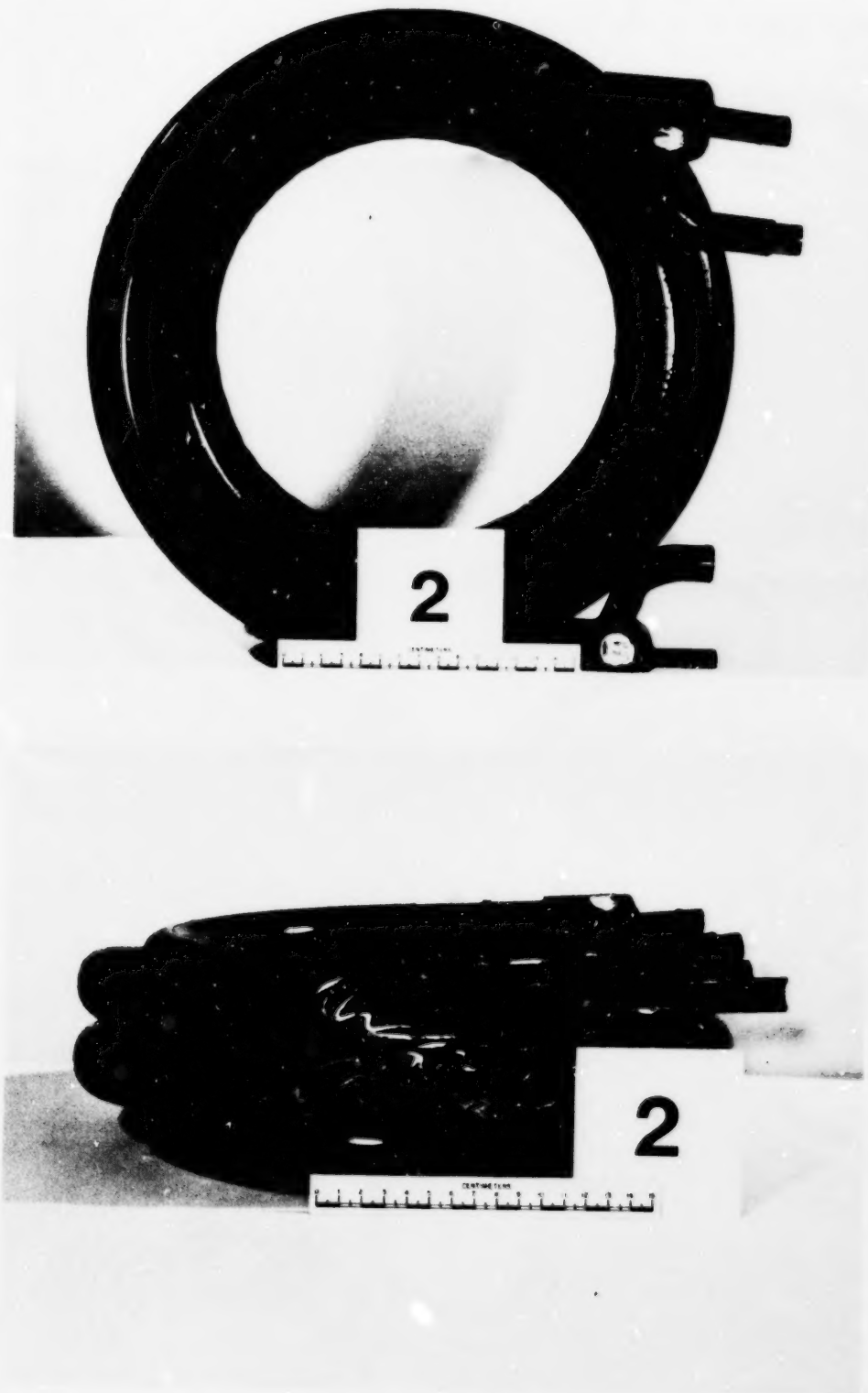
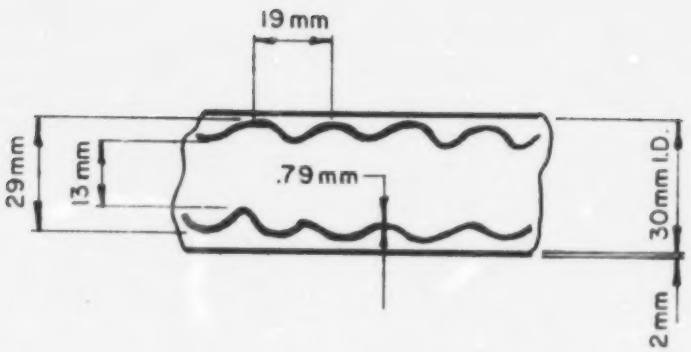


FIG. 4.2a EVAPORATOR NO. 2



APPROX. 4.7 m OF STEEL TUBE

FIG. 4.2b DETAIL OF EVAPORATOR NO. 2

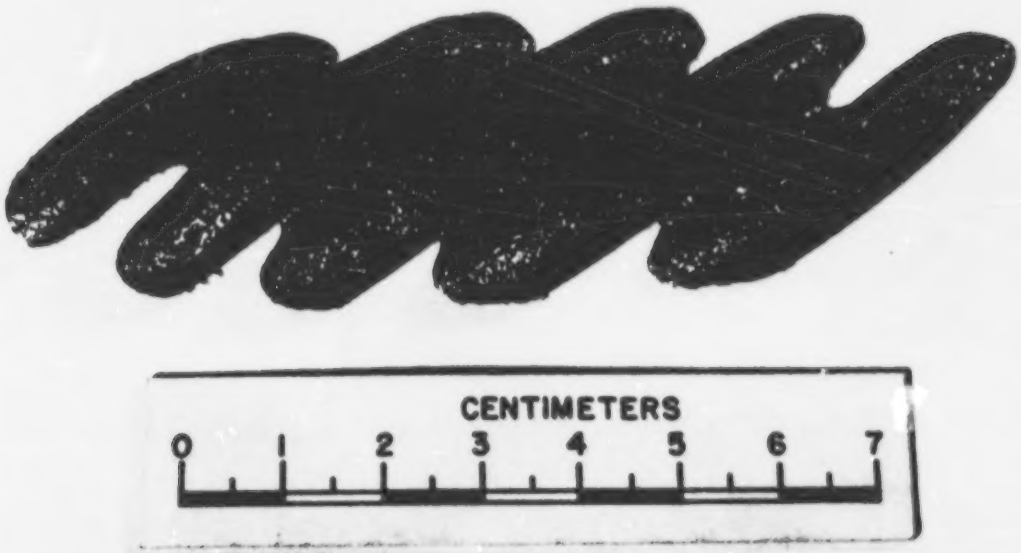
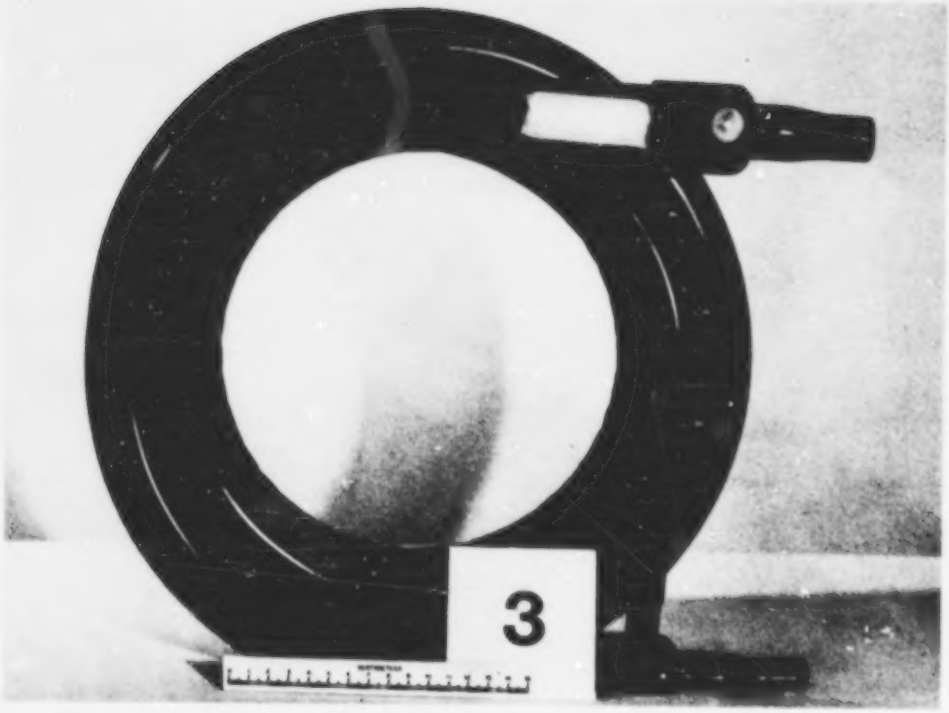


FIG. 4.2c SURFACE ENHANCEMENT OF COPPER TUBE IN EVAPORATOR NO. 2 BY SANDBLASTING



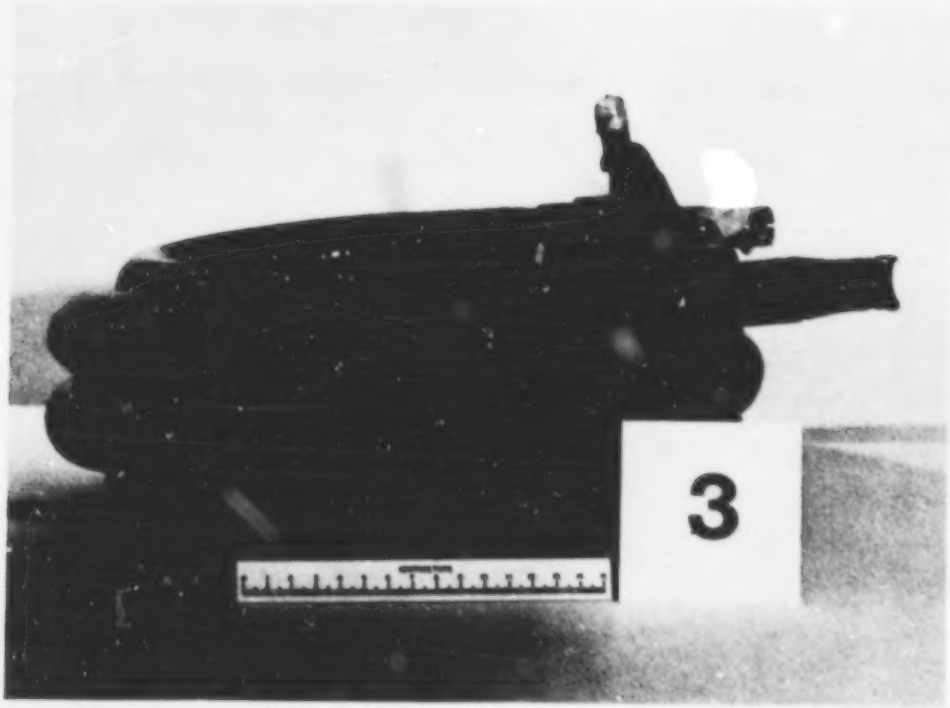
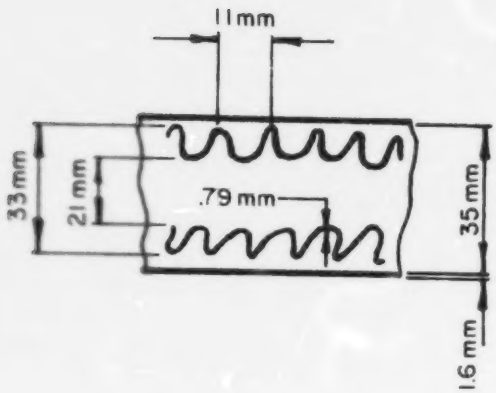
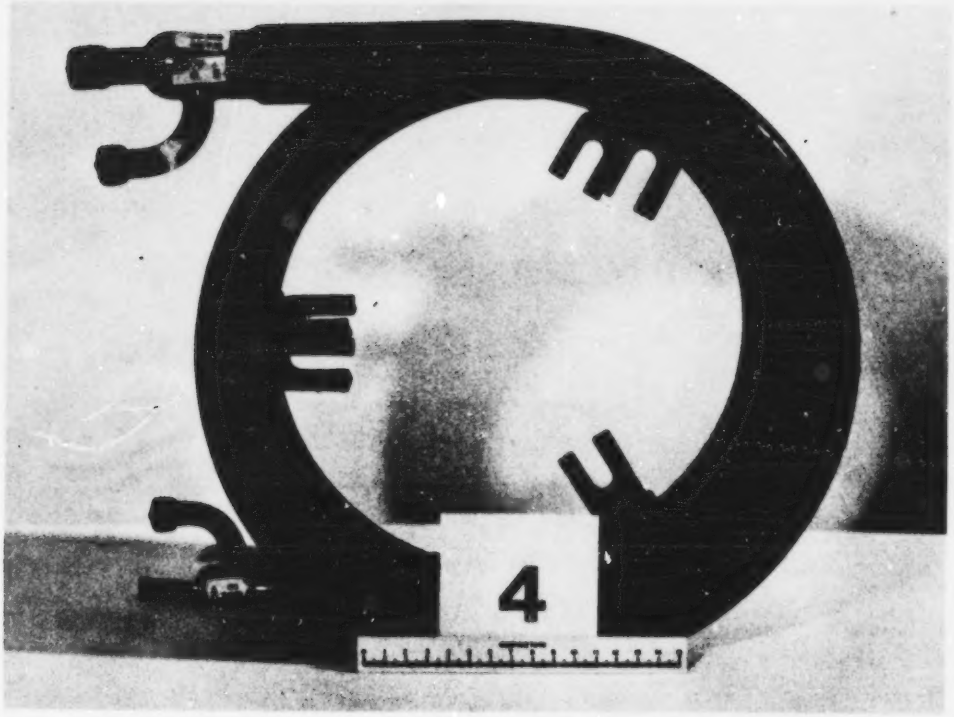


FIG. 4.3a EVAPORATOR NO. 3



APPROX. 4m OF STEEL TUBE

FIG. 4.3b DETAIL OF EVAPORATOR NO. 3



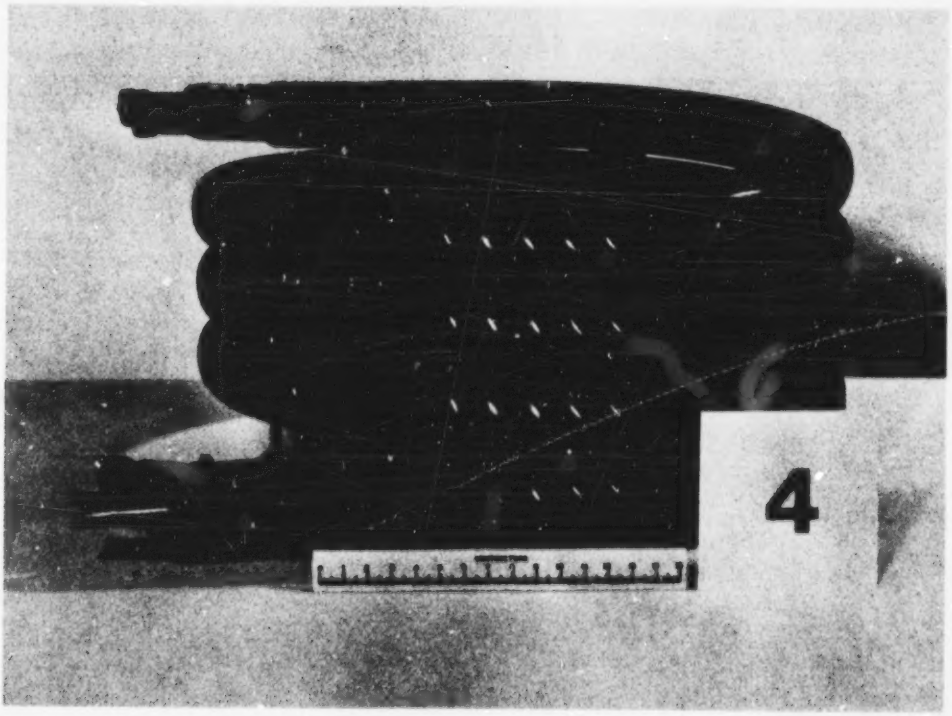


FIG. 4.4a EVAPORATOR NO. 4

[illegible] DETAIL OF EVAPORATOR NO. 4

FIG. 4.5

This figure is blank as evaporator No. 5 was a prototype on short term loan from a manufacturer. As such it could not be sectioned or shown in a Figure. The construction of evaporator No. 5 was very similar to evaporator No. 2.

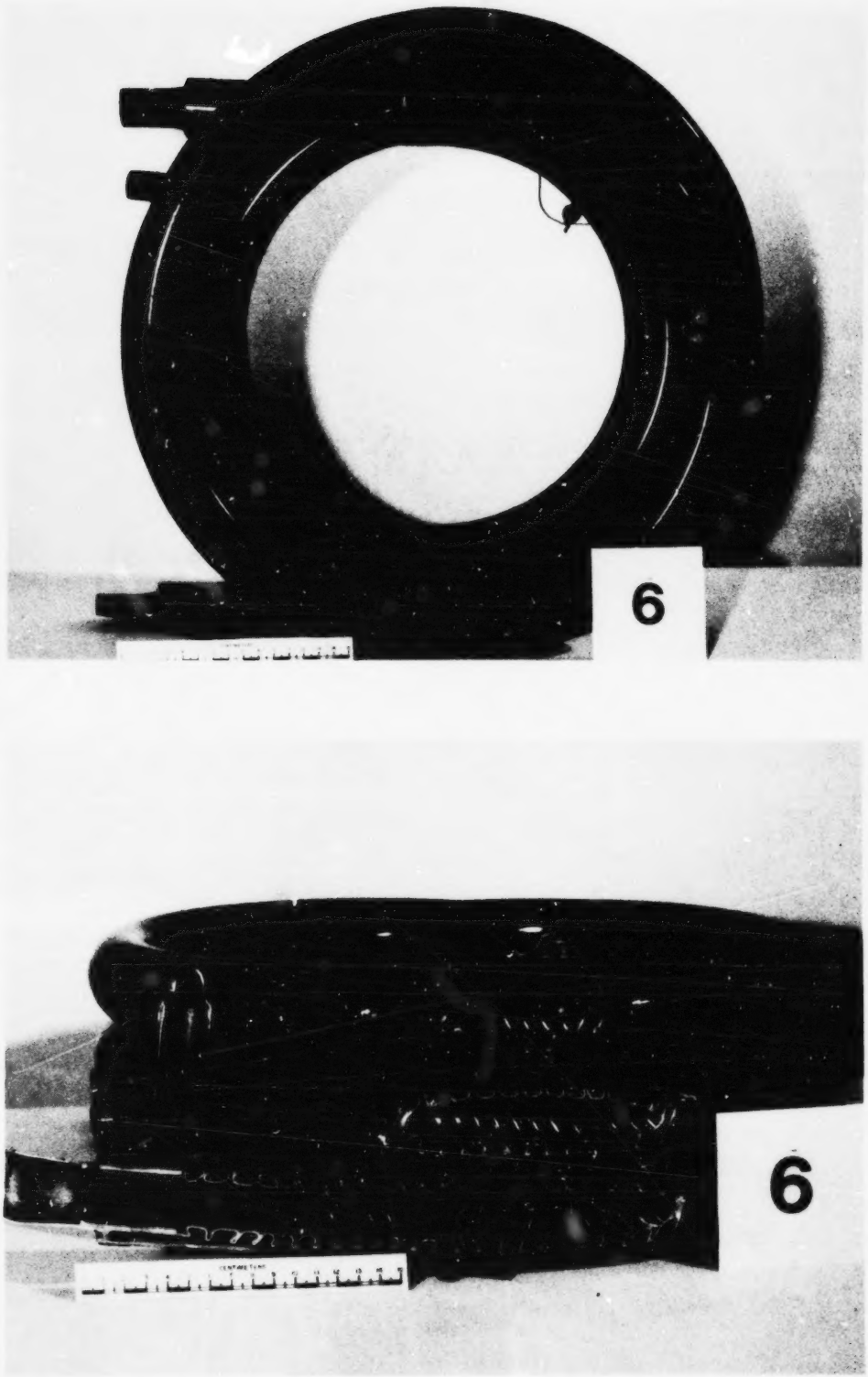
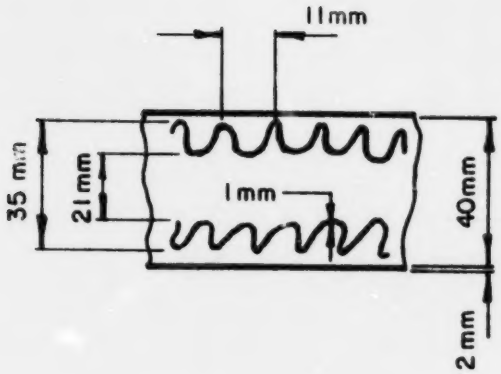


FIG. 4.6a EVAPORATOR NO. 6



APPROX. 4.9 m OF STEEL TUBE

FIG. 4.6b DETAIL OF EVAPORATOR NO. 6

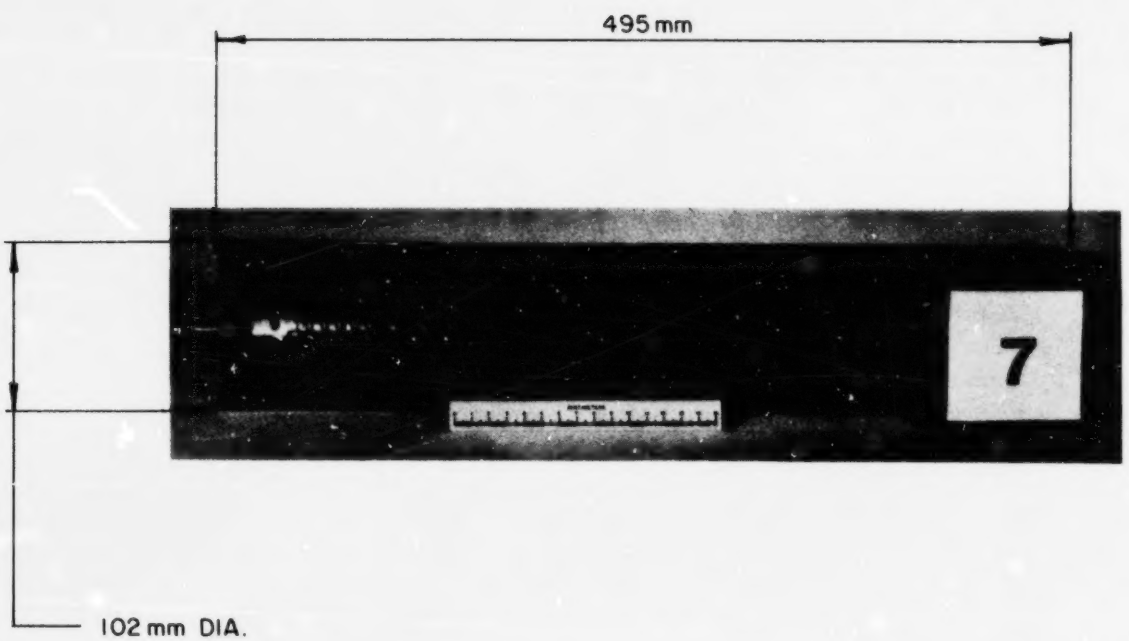


FIG. 4.7a EVAPORATOR NO. 7

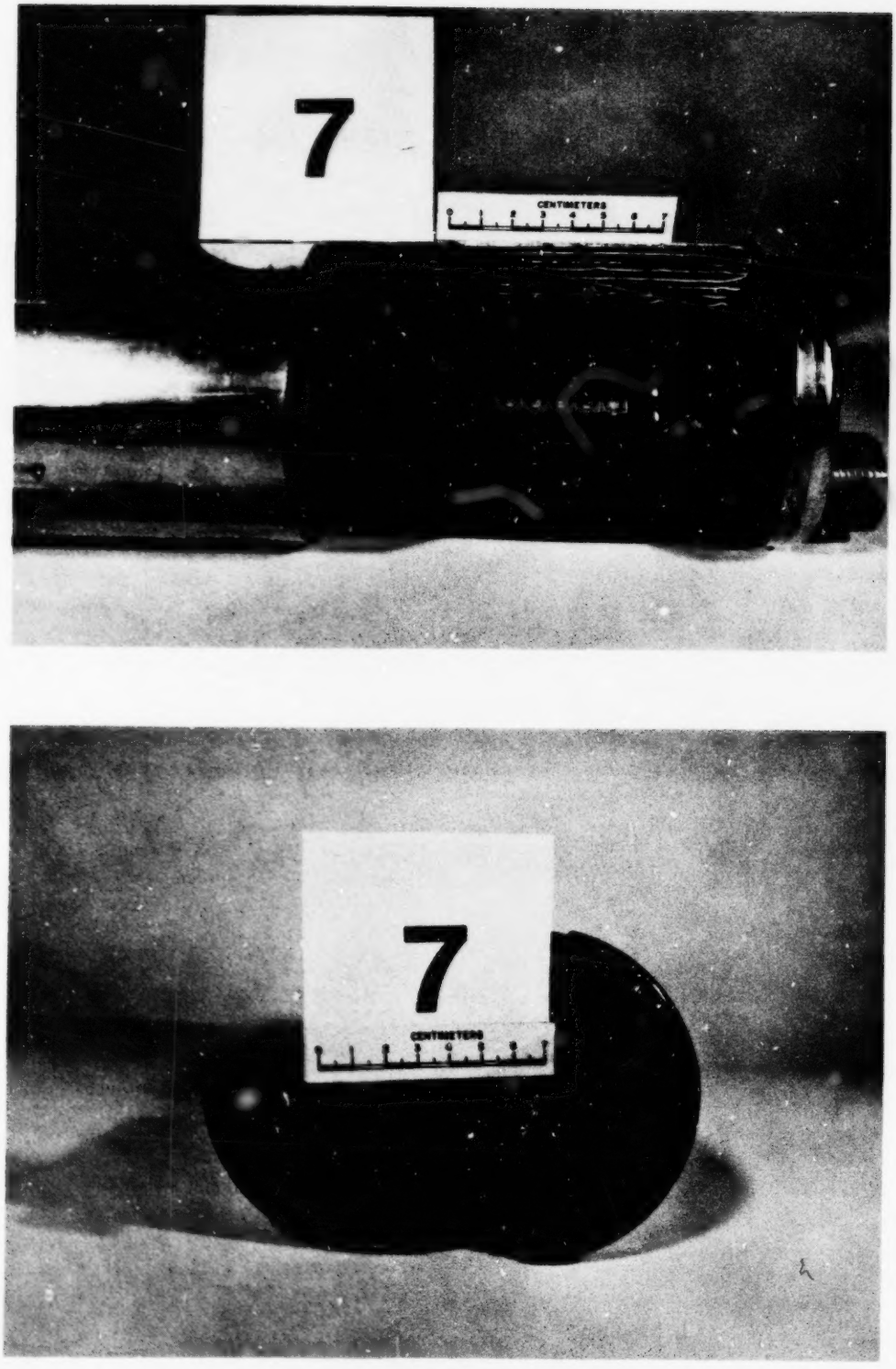
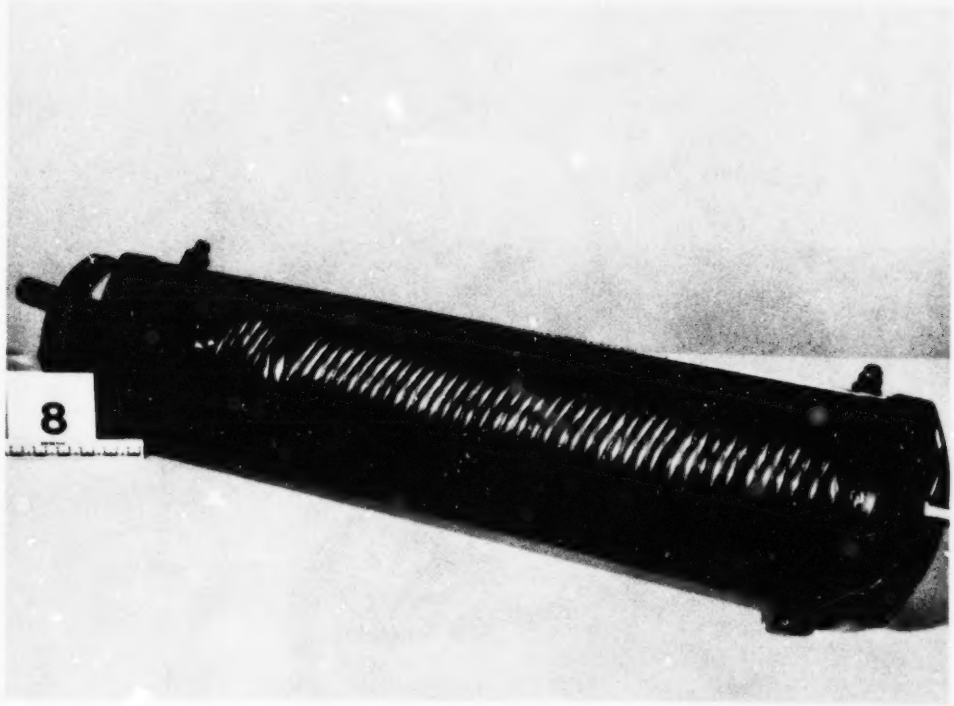


FIG. 4.7b DETAIL OF EVAPORATOR NO. 7



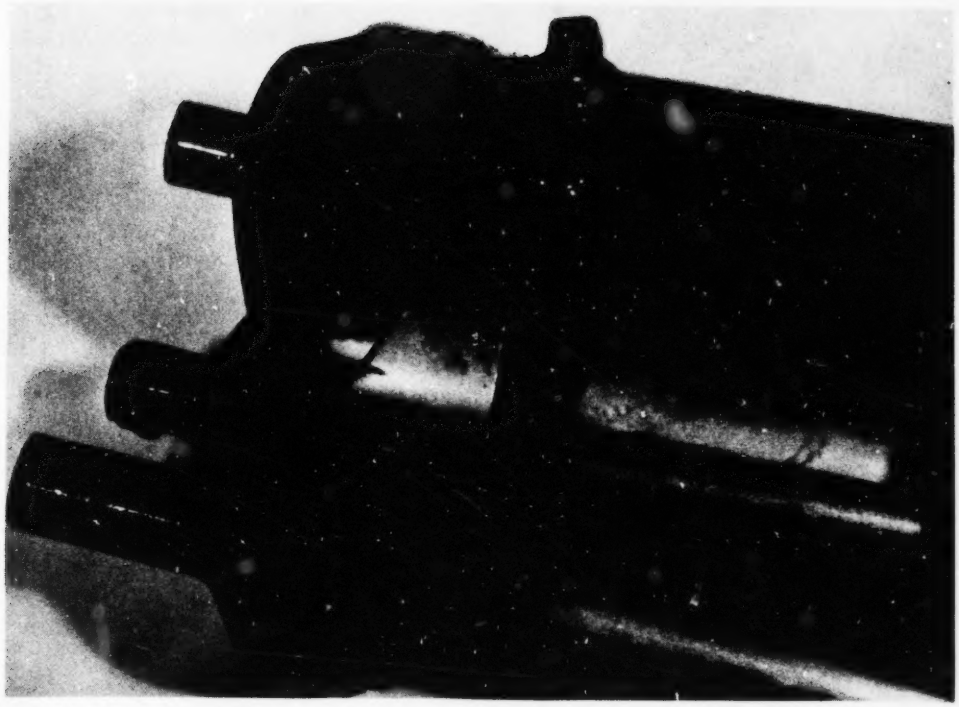


FIG. 4.8a EVAPORATOR NO. 8
FIG. 4.8b DETAIL OF TOP OF EVAPORATOR NO. 8

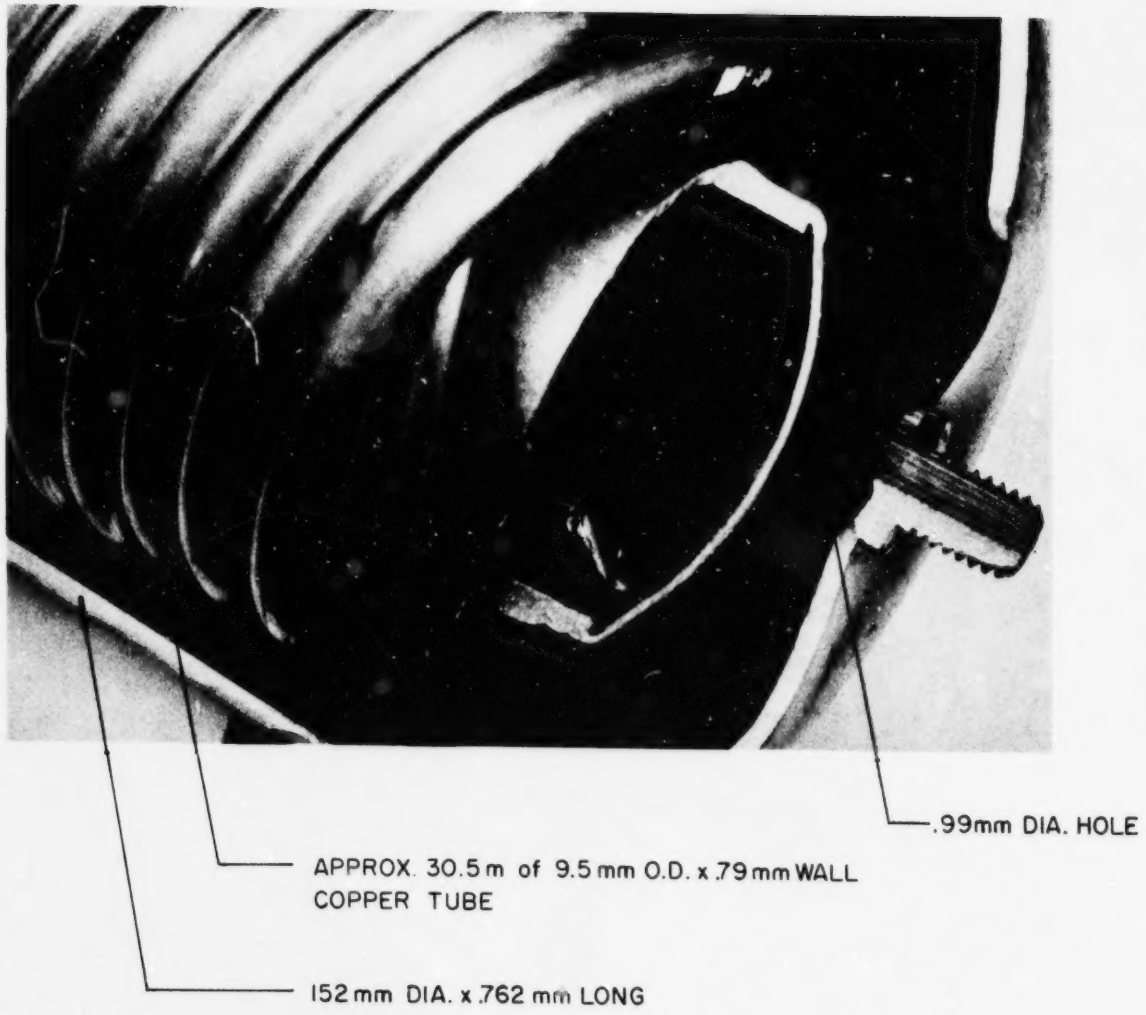


FIG. 4.8c DETAIL OF BOTTOM OF EVAPORATOR NO. 8

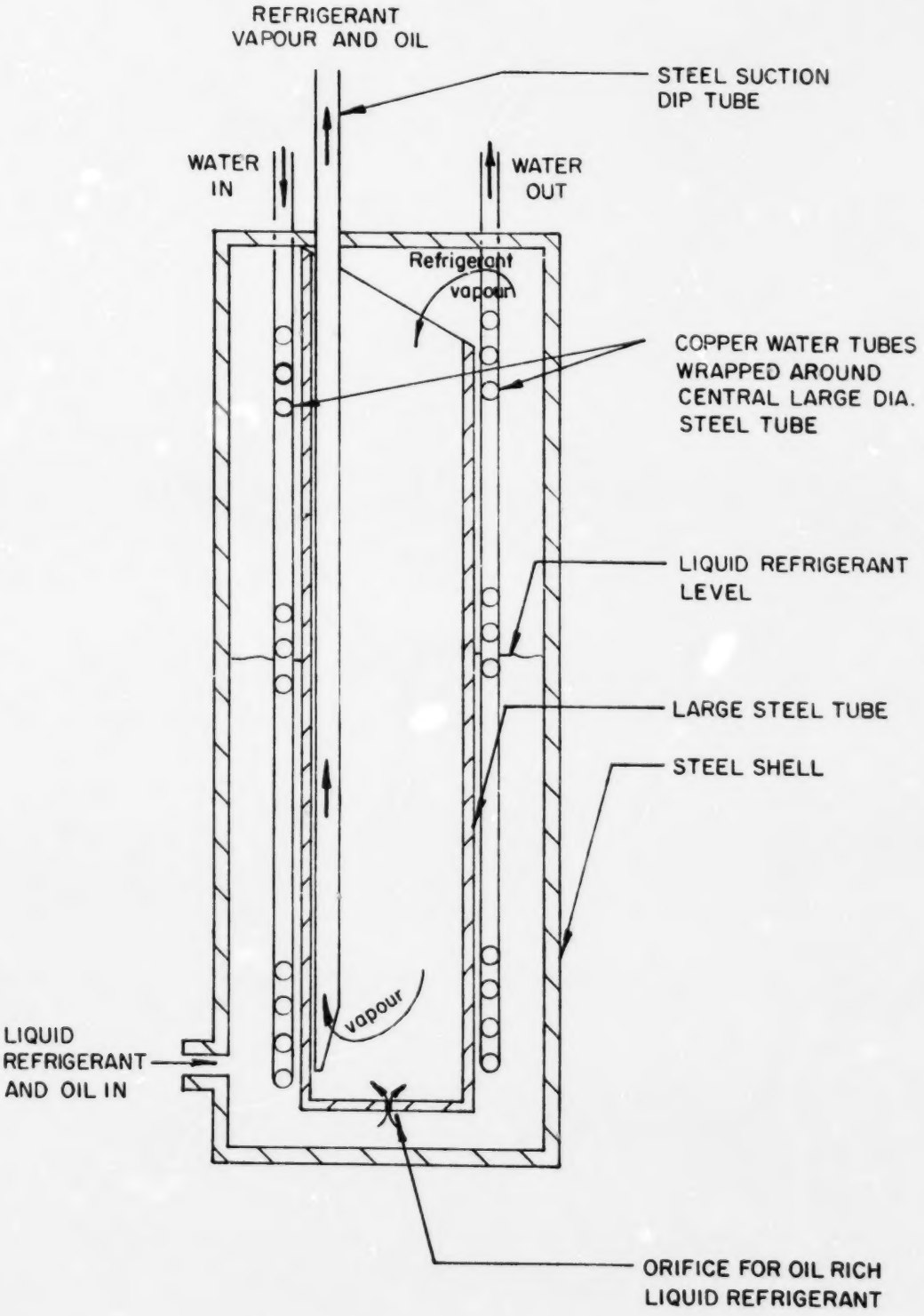


FIG. 4.8 d : EVAPORATOR 8 - PRINCIPLE OF OPERATION

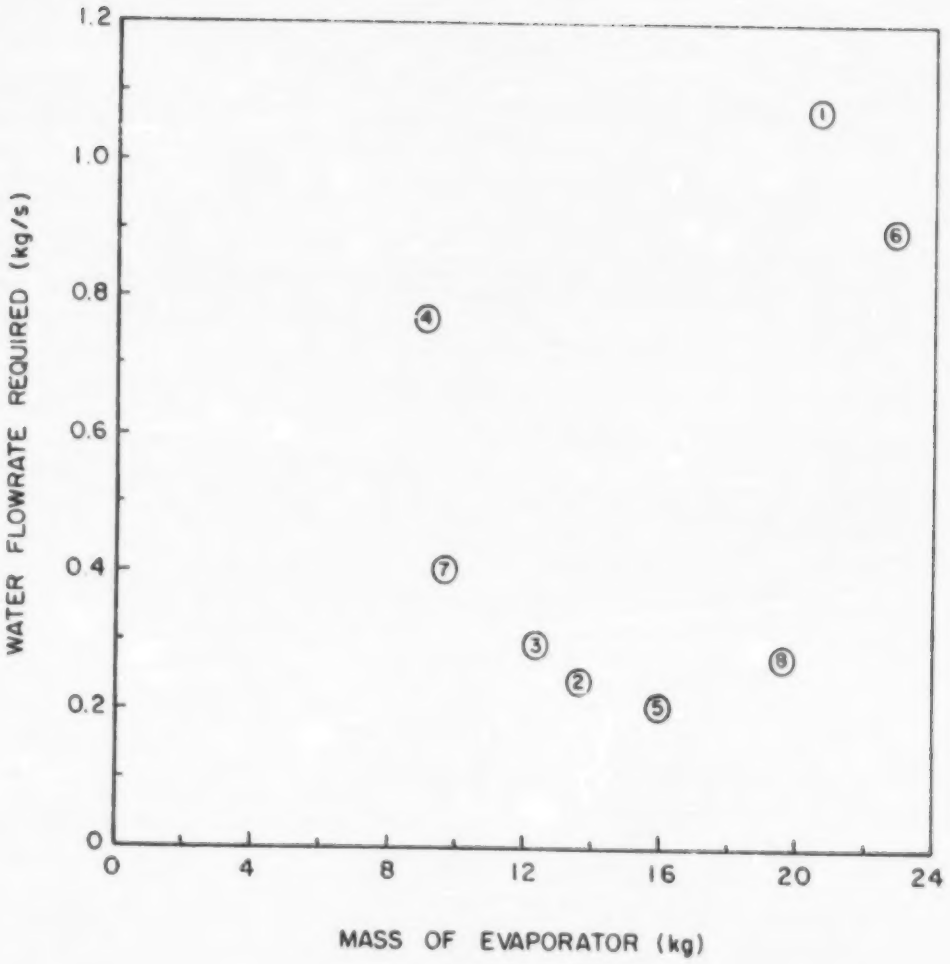


FIG. 5 WATER CONSUMPTION FOR EVAPORATORS EXTRACTING 6.6 kW FROM WATER AT 10°C USING R22 REFRIGERANT ENTERING WITH A VAPOUR MASS FRACTION OF 0.25 AND LEAVING AT A PRESSURE OF 470 kPa AND SUPERHEAT OF 5.8°C (EXCEPT FOR Nos 4,6 AND 8 WHICH USED A LOWER SUPERHEAT)

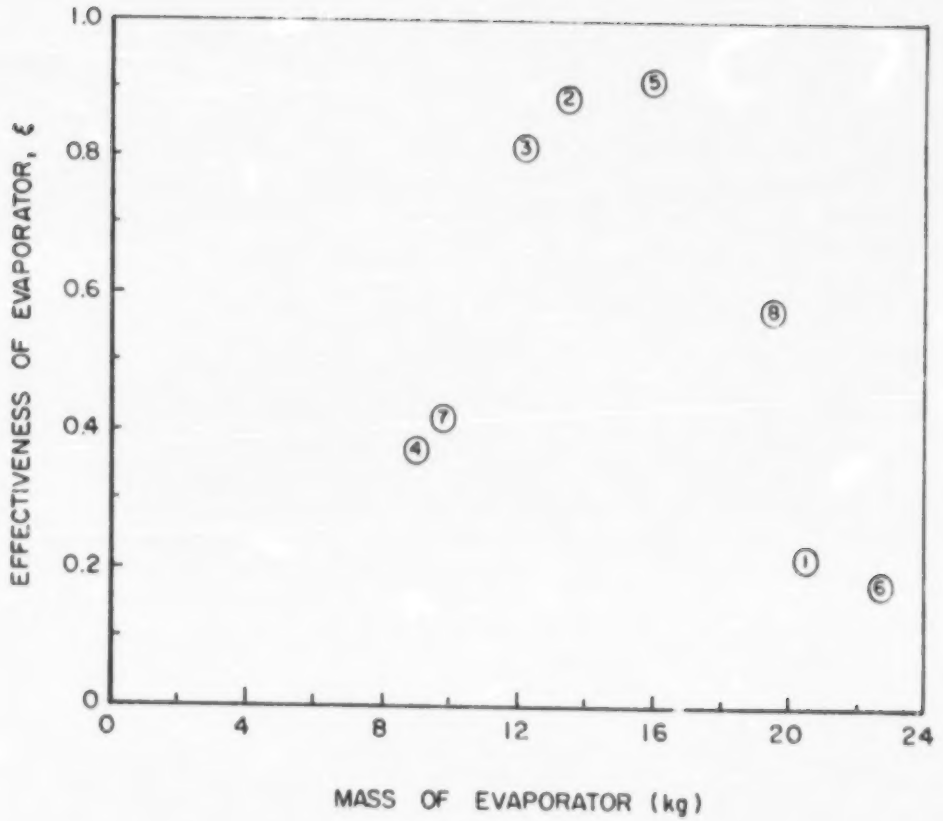


FIG. 6 EFFECTIVENESS OF EVAPORATORS EXTRACTING 6.66 kW FROM WATER AT 10 °C USING R22 REFRIGERANT ENTERING WITH A VAPOUR MASS FRACTION OF 0.25 AND LEAVING AT A PRESSURE OF 470 kPa AND SUPERHEAT OF 5.8 °C (EXCEPT FOR Nos 4,6 AND 8 WHICH USED A LOWER SUPERHEAT)

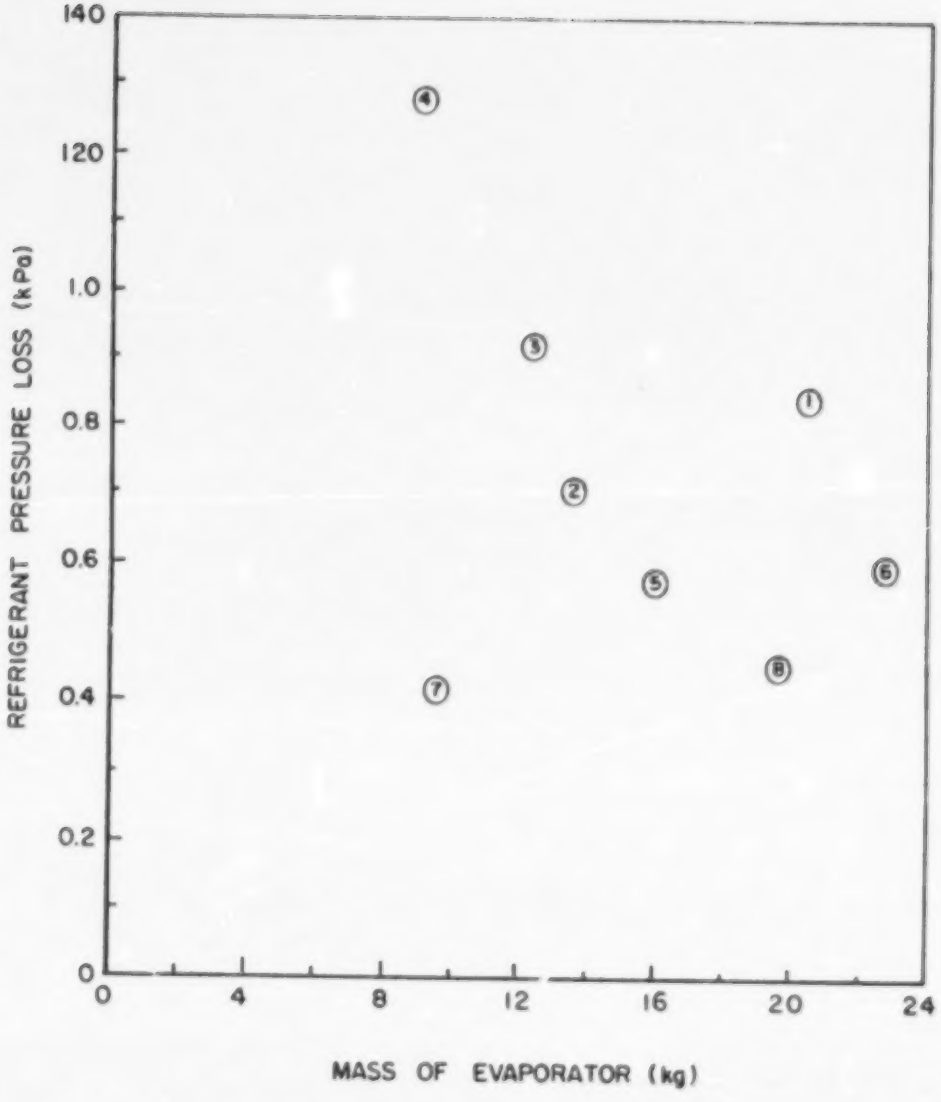


FIG. 7 REFRIGERANT CIRCUIT PRESSURE LOSS FOR EVAPORATORS EXTRACTING 6.6 kW FROM WATER AT 10°C USING R22 REFRIGERANT ENTERING WITH A VAPOUR MASS FRACTION OF 0.25 AND LEAVING AT A PRESSURE OF 470 kPa AND SUPERHEAT OF 5.8°C (EXCEPT FOR Nos 4, 6 AND 8 WHICH USED A LOWER SUPERHEAT)

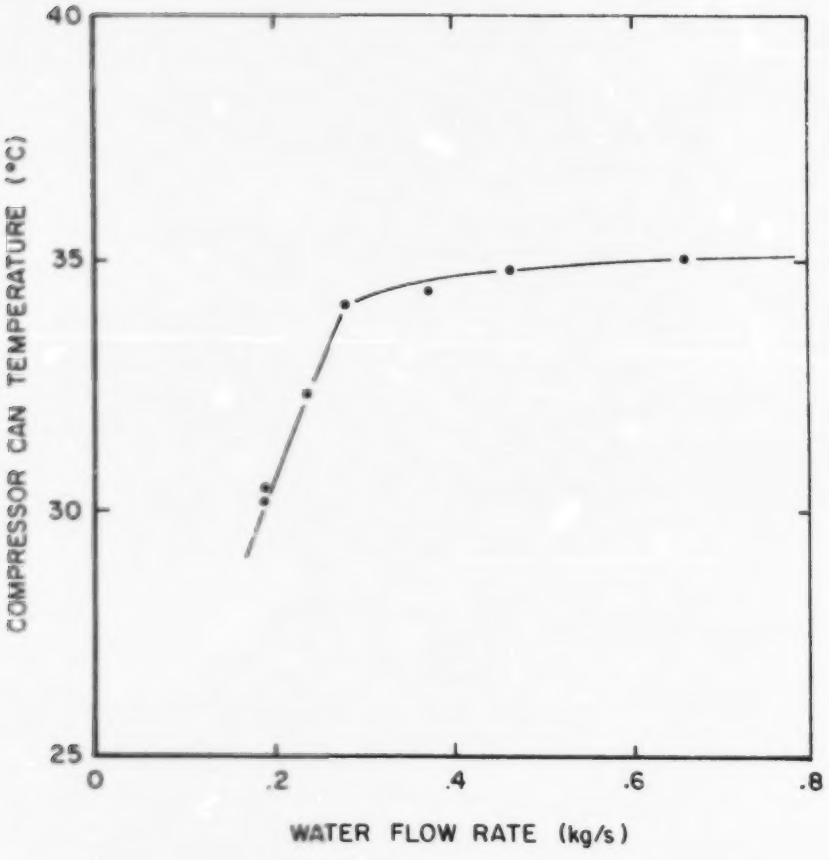


FIG. 8 EFFECT OF WATER FLOW RATE THROUGH THE 'FLOODED' TYPE EVAPORATOR (No 8) ON THE COMPRESSOR CAN TEMPERATURE. CAPACITY HELD CONSTANT AT 6.62 kW WITH A SUCTION PRESSURE OF 460 kPa AND A REFRIGERANT VAPOUR MASS FRACTION OF 0.26 AT INLET TO THE EVAPORATOR

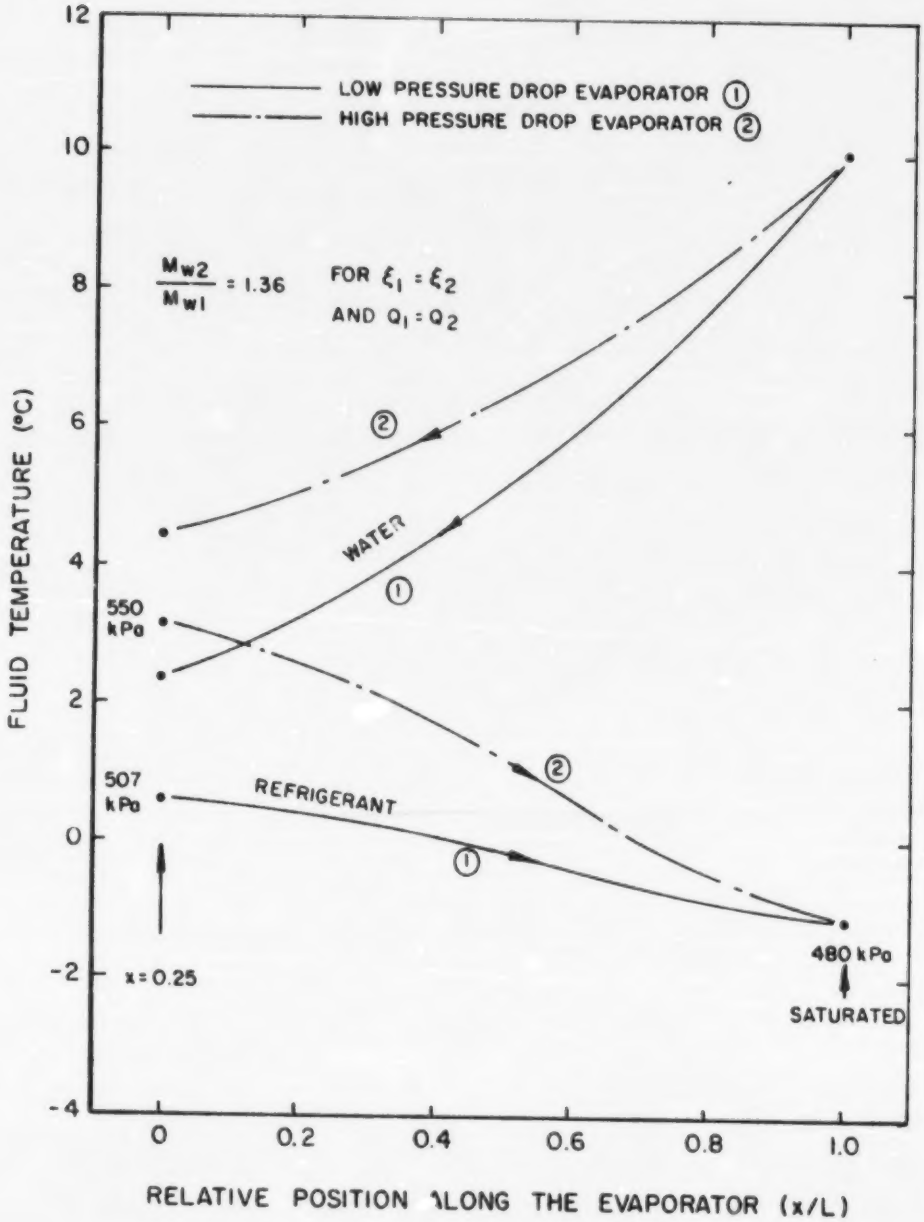


FIG. 9 EFFECT OF HIGH REFRIGERANT CIRCUIT PRESSURE DROP IN THE EVAPORATOR WITH FIXED COMPRESSOR SUCTION PRESSURE. COMPARISON ON THE BASIS OF BOTH EVAPORATORS HAVING THE SAME EFFECTIVENESS AND HEAT TRANSFER

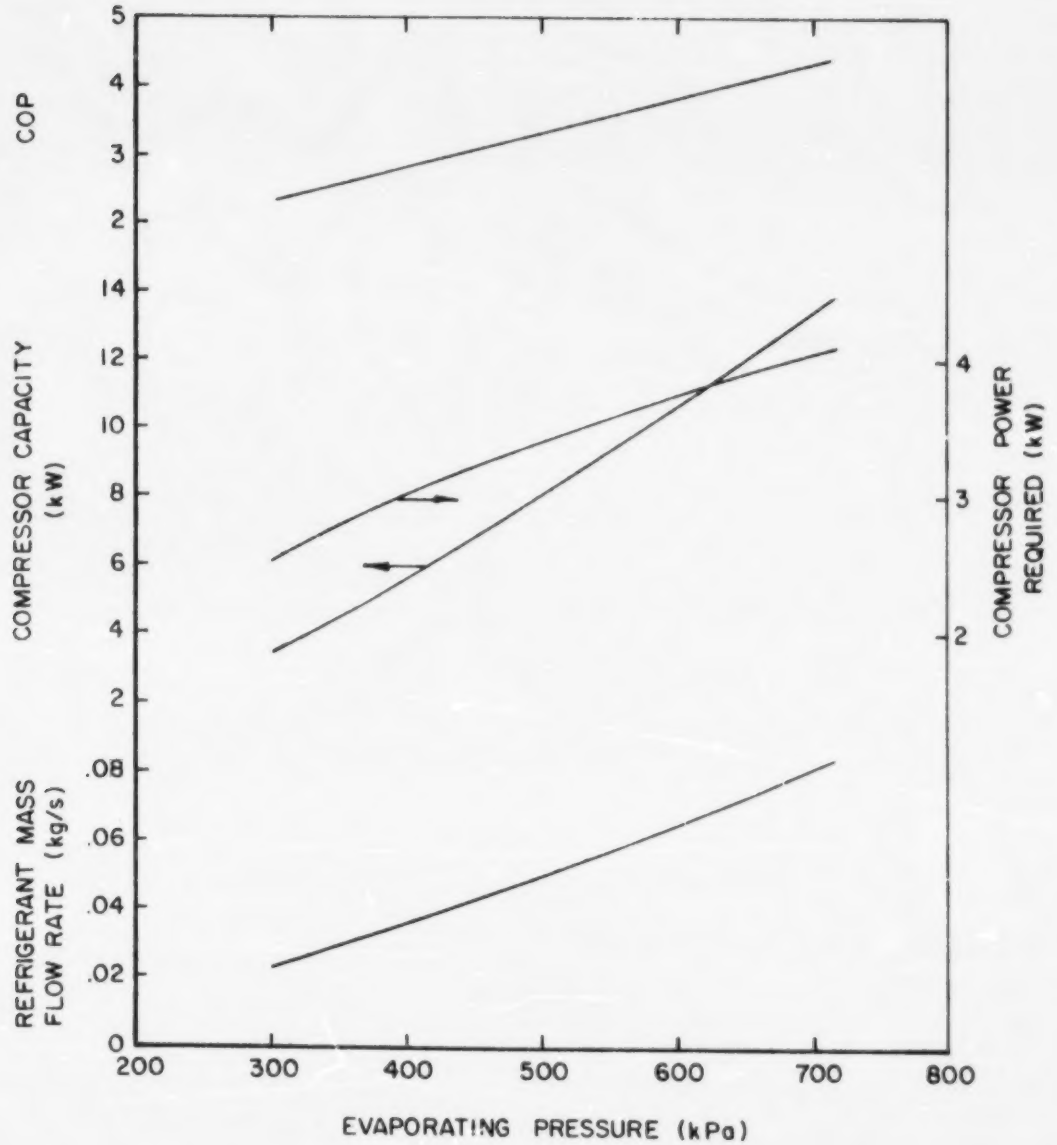
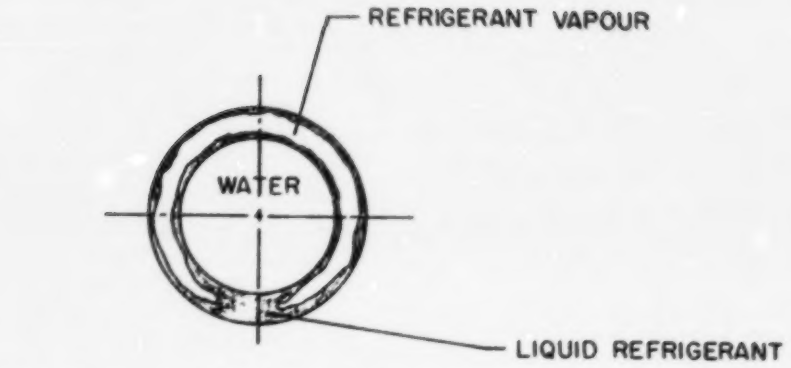
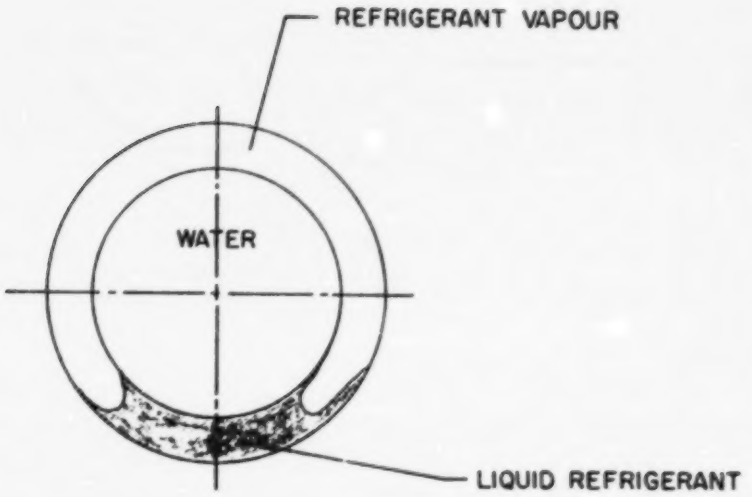


FIG. 10 EFFECT OF EVAPORATING PRESSURE ON PERFORMANCE OF A TYPICAL HERMETIC, RECIPROCATING COMPRESSOR WITH A FIXED CONDENSING PRESSURE OF 2017 kPa, 11.1°C SUPERHEATED VAPOUR TO THE COMPRESSOR AND 8.3°C SUB-COOLED LIQUID FROM THE CONDENSER



(a) CORRECTLY SIZED EVAPORATOR



(b) OVERSIZED EVAPORATOR

FIG. 11 POSTULATED REFRIGERANT FLOW PATTERN IN CORRECTLY SIZED AND OVERSIZED COAXIAL TUBE-IN TUBE EVAPORATORS

TABLE 1

DETAILS OF EVAPORATORS TESTED

Evaporator No.	Manufacturer	Type	Mass (kg)	Materials Used	Relative Cost	Relative Cost per kg
1	A	TIT	20.5	Cu,Fe	2.9	0.14
2	B	TIT	13.6	Cu,Fe	1.6	0.12
3	C	TIT	12.3	Cu,Fe	1.0	0.08
4	B	TIT	9.1	Cu,Fe	1.6	0.18
5	B	TIT	15.9	Cu,Fe	-	-
6	C	TIT	22.7	Cu,Fe	1.6	0.07
7	D	SWP	9.6	SS	3.7	0.39
8	E	TIS	19.6	Cu,Fe	2.5	0.13

Code: TIT: coaxial tube-in-tube
SWP: spiral wound plate
TIS: tube-in-shell

Cu: copper
Fe: steel
SS: stainless steel

TABLE 2

SUMMARY OF EVAPORATOR TEST RESULTS

Evaporator No.	Mass of Dry Evaporator (kg)	Refrigerant Pressure Drop (kPa)	Water Flow Required (kg/s)	Effect-iveness
1	20.5	84.1	1.077	0.22
2	13.6	71.0	0.242	0.89
3	12.3	91.7	0.295	0.81
4	9.1	128.2	0.763	0.37
5	15.9	57.9	0.204	0.91
6	22.7	60.0	0.900	0.18
7	9.6	42.1	0.398	0.42
8	19.6	45.5	0.274	0.58

TABLE 3

EFFECT OF WATER FLOW RATE ON HEAT PUMP SYSTEM PERFORMANCE

Components included in calculating the system COP	COP
Compressor only	3.27
Compressor + condenser fan	2.87
Compressor, condenser and pump for 'good' evaporator (0.204 kg/s of water required)	2.79
Compressor, condenser fan and pump for 'poor' evaporator (1.077 kg/s of water required)	2.38

APPENDIX A

TEST INSTRUMENTS USED

Function	Manufacturer	Model	Description
Expansion valve	Singer	70625-016	Electric expansion valve
Water temperature difference	Hewlett-Packard	2801A	quartz thermometer
Water temperature difference	Guildline	9536	RTD
Refrigerant flow rate	Cox	AN8-4	Turbine meter
Refrigerant flow rate	Micro-Motion	C25AF-R	Coriolis effect flow meter
Refrigerant pressures	Kavlico	P655-150A-E1A	Variable capacitance pressure transducer
Refrigerant temperatures	Thermo Electric	-	24-gauge, type T thermocouples
Data acquisition system (DAS)	Hewlett-Packard	3497A	Multi-channel data acquisition system
Computer for running DAS	Hewlett-Packard	85F	Small desktop computer

Note: The inclusion or ommission of any instrument in this summary is not intended to infer a recommendation or otherwise for that instrument.

APPENDIX B

SUMMARY OF EVAPORATOR PERFORMANCE DATA USED IN THIS REPORT

TABLE B1
Summary of data from evaporator tests - measured data

EVAPORATOR No.		1	2	3	4	5
TEST DATE		260784	70984	100884	150884	60984
WATER CIRCUIT:						
Flow rate	(kg/s)	1.077	.242	.295	.763	.204
Temp in	(C)	10.33	10.05	10.04	10.45	10.00
Temp out	(C)	8.86	3.41	4.68	8.56	2.32
Pressure loss	(kPa)	34.5	15.9	9.0	100.0	5.5
REFRIGERANT CIRCUIT:						
Flow rate	(kg/s)	0.04324	0.04399	0.04392	0.04097	0.04369
Temp of sub-cooled liq to exp valve	(C)	43.5	42.6	43.3	42.8	43.3
Press of sub-cooled liq to X valve	(kPa)	2183	1991	1922	2214	1972
Temp into evap	(C)	3.7	2.6	3.4	5.3	1.6
Press in	(kPa)	563.9	545.3	563.9	595.0	526.7
Temp out of evap	(C)	4.8	4.3	4.1	.3	4.1
Press out	(kPa)	479.8	474.3	472.2	466.7	468.8

EVAPORATOR No.		6	7	8	9	10
TEST DATE		110984	120984	140984	0	0
WATER CIRCUIT:						
Flow rate	(kg/s)	.900	.398	.274	0.000	0.000
Temp in	(C)	10.11	10.07	10.10	0.00	0.00
Temp out	(C)	8.40	6.02	4.32	0.00	0.00
Pressure loss	(kPa)	60.0	4.1	24.1	0.0	0.0
REFRIGERANT CIRCUIT:						
Flow rate	(kg/s)	0.04248	0.04407	0.04475	0.00000	0.00000
Temp of sub-cooled liq to exp valve	(C)	41.6	43.1	43.1	0.0	0.0
Press of sub-cooled liq to X valve	(kPa)	1972	1893	1858	0	0
Temp into evap	(C)	.6	.4	.2	0.0	0.0
Press in	(kPa)	514.3	510.2	504.6	0.0	0.0
Temp out of evap	(C)	-1.9	3.7	-1.6	0.0	0.0
Press out	(kPa)	454.3	468.1	459.1	0.0	0.0

TABLE B2
Summary of data from evaporator tests - calculated data

EVAPORATOR No.	1	2	3	4	5
WATER CIRCUIT:					
Heat lost (W)	6636	6730	6622	6047	6567
REFRIGERANT CIRCUIT:					
Sub-cooling of liq before exp valve (C)	11.7	8.5	6.2	13.0	7.4
Quality of refrig into evaporator	0.25	0.25	0.25	0.23	0.26
Enthalpy in (kJ/kg)	99.0	97.8	98.7	98.0	98.7
Super heat (C)	5.9	5.8	5.7	2.2	5.9
Enthalpy out (kJ/kg)	253.8	253.6	253.4	250.8	253.5
Pressure drop (kPa)	84.1	71.0	91.7	128.2	57.9
Heat gained (W)	6694	6855	6796	6263	6767
Error in energy balance (%)	0.9	1.9	2.6	3.6	3.0
Effectiveness	0.22	0.89	0.81	0.37	0.91

EVAPORATOR No.	6	7	8	9	10
WATER CIRCUIT:					
Heat lost (W)	6435	6752	6631	0	0
REFRIGERANT CIRCUIT:					
Sub-cooling of liq before exp valve (C)	9.0	5.8	5.0	0.0	0.0
Quality of refrig into evaporator	0.25	0.26	0.26	0.00	0.00
Enthalpy in (kJ/kg)	96.4	98.4	98.4	0.0	0.0
Super heat (C)	.8	5.6	.8	0.0	0.0
Enthalpy out (kJ/kg)	249.5	253.3	249.6	0.0	0.0
Pressure drop (kPa)	60.0	42.1	45.5	0.0	0.0
Heat gained (W)	6504	6824	6771	0	0
Error in energy balance (%)	1.1	1.1	2.1	0.0	0.0
Effectiveness	0.18	0.42	0.58	0.00	0.00

TABLE B3
Summary of data from evaporator tests - measured data

EVAPORATOR No.		2H	2V
TEST DATE		70984	171084
WATER CIRCUIT:			
Flow rate	(kg/s)	.242	.221
Temp in	(C)	10.05	10.05
Temp out	(C)	3.41	2.91
Pressure loss	(kPa)	15.9	12.4
REFRIGERANT CIRCUIT:			
Flow rate	(kg/s)	0.04399	0.04369
Temp of sub-cooled liq to exp valve	(C)	42.6	43.0
Press of sub-cooled liq to X valve	(kPa)	1991	1948
Temp into evap	(C)	2.6	2.3
Press in	(kPa)	545.3	549.5
Temp out of evap	(C)	4.3	3.9
Press out	(kPa)	474.3	474.3

TABLE B4
Summary of data from evaporator tests - calculated data

EVAPORATOR No.		2H	2V
WATER CIRCUIT:			
Heat lost	(W)	6720	6627
REFRIGERANT CIRCUIT:			
Sub-cooling of liq before exp valve	(C)	8.5	7.1
Quality of refrig into evaporator		0.25	0.25
Enthalpy in	(kJ/kg)	97.8	98.3
Super heat	(C)	5.8	5.4
Enthalpy out	(kJ/kg)	253.6	253.3
Pressure drop	(kPa)	71.0	75.1
Heat gained	(W)	6855	6773
Error in energy balance	(%)	1.9	2.2
Effectiveness		0.89	0.93

APPENDIX C

EFFECT OF WATER FLOW RATE ON HEAT PUMP SYSTEM PERFORMANCE

Assumptions:

1.	Typical 10.5 kW water source heat pump using R22	
2.	Evaporating temp. T_e	-1.1°C
3.	Condensing temp, T_c	51.7°C
4.	Superheat of vapour to compressor, T_{sh}	5.6°C
5.	Sub-cooling of liquid refrigerant from condenser, T_{sc}	8.3°C
6.	Condenser airflow, V_{cf}	566 l/s
7.	Pressure loss seen by condenser fan, ΔP_{cf}	224 Pa
8.	Efficiency for conversion of electricity to air power, E_{cf}	0.2
9.	Water temperature in well, T_w	10°C
10.	Static head for water pump, H_s	30 m
11.	Friction head factor, K	10.0
12.	Efficiency for conversion of electricity to water power, E_{wp}	0.60

Power required by condenser fan, P_c, is given by:

$$P_c = \frac{V_{cf}\ \Delta P_{cf}}{1000\ E_{cf}} \qquad (C1)$$

i.e. $P_c = 630W$

The evaporators were tested at a common capacity, Q_e, of 6.66 kW. From the manufacturer's data sheets, a typical hermetic, reciprocating compressor of the above size will have a coefficient of performance (COP) of 3.27 at the above conditions.

For the compressor only (neglecting losses):

$$COP_c = \frac{Q_e + P_c}{P_c} \qquad (C2)$$

where P_c is the power required by the compressor.

Therefore: $P_c = 2.93$ kW

The friction head of the water system, H_f, is given by

$$H_f = KV_w^2 \qquad (C3)$$

where V_w is the volumetric flow rate of water (1/s = kg/s).

The power required by the water pump, P_{wp}, is given by:

$$P_{wp} = \frac{g\rho\,(H_s + H_f)\,V_w}{1000\;E_{wp}} \qquad (C4)$$

where ρ is the density of the water and g is the gravitational acceleration (9.8 m/s^2).

Pump power required for 'good' and 'poor' evaporators:

V_w (l/s)	$(H_s + H_f)$ (m)	P_{wp} (kW)
0.204	30.42	0.102
1.077	41.60	0.732

System Coefficient of Performance (COP_s) is defined as:

$$COP_s = \frac{\text{energy delivered}}{\text{electrical energy required}}$$

$$\simeq \frac{P_c + P_{cf} + Q_e}{P_c + P_{cf} + P_{wp}} \qquad (C5)$$

Note that the small amount of power consumed by the heat pump controls has been neglected.

REPORT DOCUMENTATION PAGE/PAGE DE DOCUMENTATION DE RAPPORT

REPORT/RAPPORT	REPORT/RAPPORT
TR-LT-010 (1a)	NRCC 24201 (1b)
REPORT SECURITY CLASSIFICATION / CLASSIFICATION DE SÉCURITÉ DE RAPPORT: UNCLASSIFIED (2)	DISTRIBUTION/DIFFUSION: UNLIMITED (3)

TITLE/SUBTITLE/TITRE/SOUS-TITRE (CLASSIFICATION) Unclassified (4)

TESTING OF WATER-TO-REFRIGERANT EVAPORATORS FOR RESIDENTIAL HEAT PUMP SYSTEMS

AUTHOR(S)/AUTEUR(S) (5)

Hawken, P. J.; Hearty, P. F.

SERIES/SÉRIE (6)

Technical Report

CORPORATE AUTHOR/PERFORMING AGENCY/AUTEUR D'ENTREPRISE/AGENCE D'EXÉCUTION (7)

National Research Council, Low Temperature Laboratory
Division of Mechanical Engineering

SPONSORING AGENCY/AGENCE DE SUBVENTION (8)

DATE (9)	FILE/DOSSIER (10)	LAB. ORDER / COMMANDE DU LAB. (11)	PAGES (12a)	DIAGS (12b)	REFS (12c)
1984/11			53	11	3

NOTES (13)

DESCRIPTORS(KEY WORDS)/MOTS-CLÉS (14)

Evaporator, Heat Pump, Testing

SUMMARY/SOMMAIRE (15)

Tests were made of eight commercially available, water-to-refrigerant evaporators suitable for residential heat pumps. The effect of evaporator performance on the heat pump system is discussed.

ADDRESS/ADRESSE (16)

T. R. Ringer, Head,
Low Temperature Laboratory,
Division of Mechanical Engineering,
National Research Council of Canada,
Montreal Road, Ottawa, Ontario.
K1A 0R6 (613) 993-2439